POWER

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For Example

On taking charge of a saw-mill plant having one 60 in.x12 ft. boiler and a 12x18 in. slide-valve engine, the owner assured me that everything was in good condition. I soon found, however, that the steam had been down for a half hour out of every two for a month. On Sunday I began work by "punching" soot from the flues and cleaning the combustion chamber, and at 5 a. m. Monday I put in my first fire. I used the mill refuse for fuel, and in an hour had 95 lb. of steam.

I made up my mind that the fault was not all in the boiler, but I had been too busy the day before to give attention to the engine. When I started it I wished that I had, as the sound of the exhaust was exceedingly loud and every joint seemed loose. When the sawyers stalled the engine by making a large cut I promptly shut down and refused to start again until I had an opportunity to overhaul it. This angered the foreman and, to my surprise, the owner also. He finally said: "You can do as you like, about it, but you can't spend a cent for new material."

With a bar I could raise the crosshead $\frac{3}{8}$ in. in the guides; the pillow block had been hot and melted part of the babbitt; the flywheel was loose on the shaft, and the brasses had been severely burned. I scraped and babbitted the brasses and pillow block, tightened the guides and set the valve, but on starting, the engine was almost as noisy as before.

Then I took off the cylinder head and found the piston loose on the rod, and after much arguing I was allowed to get a new rod. I thought my troubles were over, but on starting again the pounding was still very severe. I calipered the pins and found them flat, and told the owner I must have new ones, but he refused to buy them.

The 4-in. exhaust pipe ran out through the side of the mill, but instead of using an elbow to deflect the steam to the ground, a bend was made, the inside of which buckled, so that there was only a $\frac{3}{4}$x2 in. opening.

As the management would not support me in my efforts to put the plant in running order, I left. I had several successors, but eventually central-station service was used. They still have to keep steam on the boiler seven months in the year for steaming cheesebox hoops and two or three days a week they have to carry 100 lb. All this requires the service of a fireman. The surplus sawdust is allowed to run into the river and is an absolute waste of good fuel.

Thirty dollars would have put that engine in good condition, and with fuel costing practically nothing, it is evident how much the owner could have saved by running his own plant. Under the circumstances it would seem as though the engineer's advice should have been given a little more consideration.

This letter, recently received, recounts an experience that's regrettably common. It is one of the many instances where an ignorant owner had an intelligent engineer—several of them, in fact—but rather than take his engineer's advice he purchased central-station current, thereby increasing his power costs and wasting fuel which cost him practically nothing.

Unfortunately, the reasons for changing over are not always made known to the public and the engineer of the isolated plant is blamed. This letter should convince the hesitant owner that the engineer usually knows best how to operate the plant and should be consulted.

Coal and Ash Handling at Sioux City

A complete system for the power plant with facilities for saving the small coal which drops through the grate and returning it with the raw coal to the furnace.

The plant of the Sioux City Service Co., at Sioux City, Iowa, has been partially rebuilt and equipped with an unusually complete system for handling coal and ashes. The boiler and engine rooms are side by side under the same roof. The former is about 77 ft. wide and 131 ft. long. The boilers are in two rows along the side walls and face toward the center of the room. A large serving space between the boilers allows withdrawing the chain-grate stokers. The boilers, of Stirling type in 400-hp. units, total 3300 hp. The stack is at one side of the room and is central to the boilers. On the other side, directly opposite the stack, is the water tower.

The railroad track, on which coal is received and to which ashes are delivered, runs past the ends of the engine and boiler rooms, thus coming in at right angles to the conveyor system, which is an ideal arrangement. Provision has been made for storing a certain amount of coal on the ground outside of the building which may be used in an emergency, but the main storage of 850 tons is in steel bins above and midway between the two lines of boilers. The coal from the bins is fed through valves and chutes directly to the stoker hoppers and no hand labor is required.

To provide storage for ashes, the conveyor system is continued over the coal track and an ash storage of 140 tons is afforded by a steel bin just outside the boiler room. The ashes from this bin pass through valves and chutes to the same car in which the coal is received.

One of the features of the system is the saving of small particles of coal from the chain grates. In some installations this fine coal has been reclaimed by shoveling it back into the stokers from a small receiving pit in the floor, but more often it has gone into the ashes. In the present case hoppers have been so arranged underneath the stokers that they receive practically all of the fine coal which is of value and yet allow space at the end of the grates for the ashes to fall into the pit beneath. When a quantity of coal has accumulated in the hoppers, it is fed slowly into the conveyor with the raw coal from the receiving hopper. This mixes the finer particles of coal dust and partially burned coal with the raw coal and enables the former to be used advantageously in the stokers.

The coal- and ash-handling system, shown in outline in Fig. 1, consists of a pivoted bucket carrier which runs underneath the coal-receiving hopper and through the basement, thence ascending to the upper runway, discharging the coal to the coal bins and the ashes to the ash bin and descending outside of the building to the receiving hopper.

The coal from the cars is dumped into the receiving hopper and then fed by a reciprocating plate, or feeder, to a crusher underneath the feeder, or is bypassed to a chute beyond the crusher and delivered directly to the conveyor. This avoids running the crusher when slack coal is available. The power for operating the feeder is furnished by the conveyor itself. The arrangement of the feeder and crusher is shown in Fig. 4. The opposite

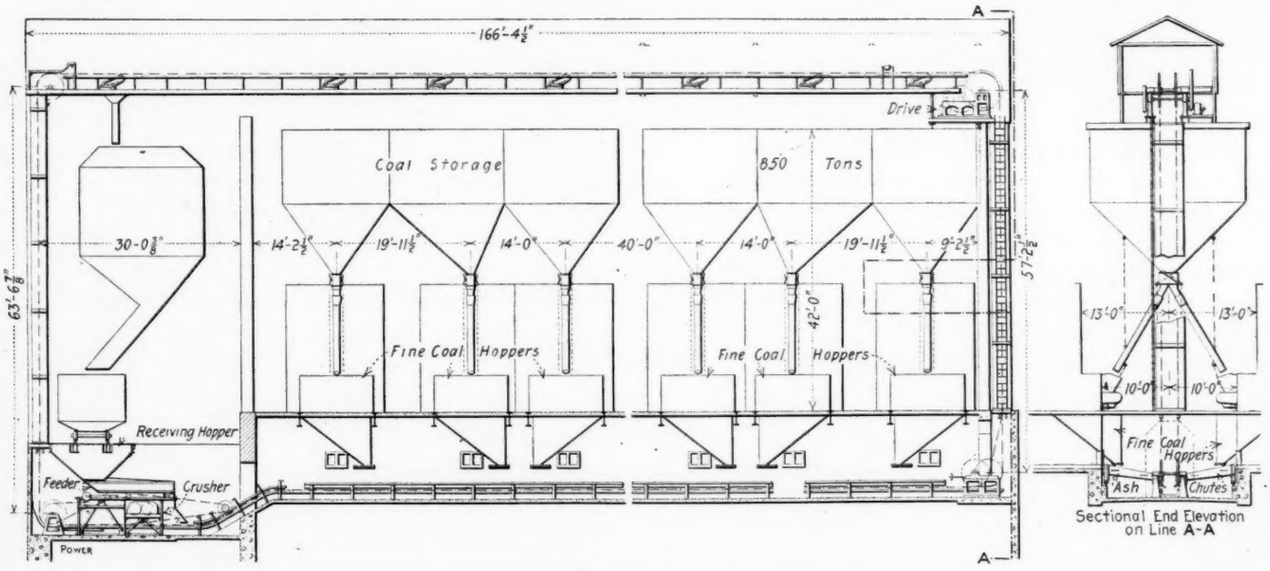


FIG. 1. SECTIONAL ELEVATIONS OF BOILER ROOM, SHOWING CONVEYING SYSTEM

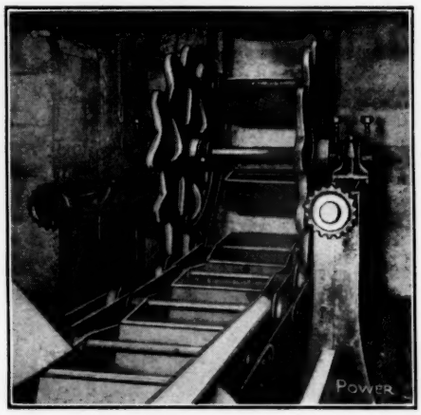


FIG. 2. END OF LOWER RUN OPPOSITE CRUSHER

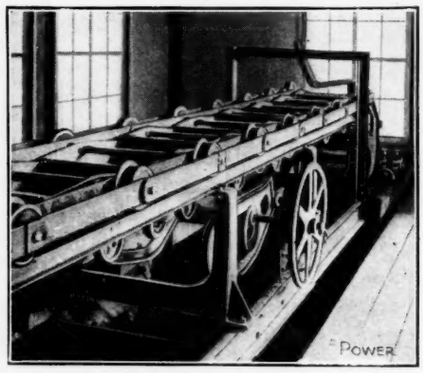


FIG. 3. UPPER RUN, SHOWING TRIPPER OVER ASH CHUTE

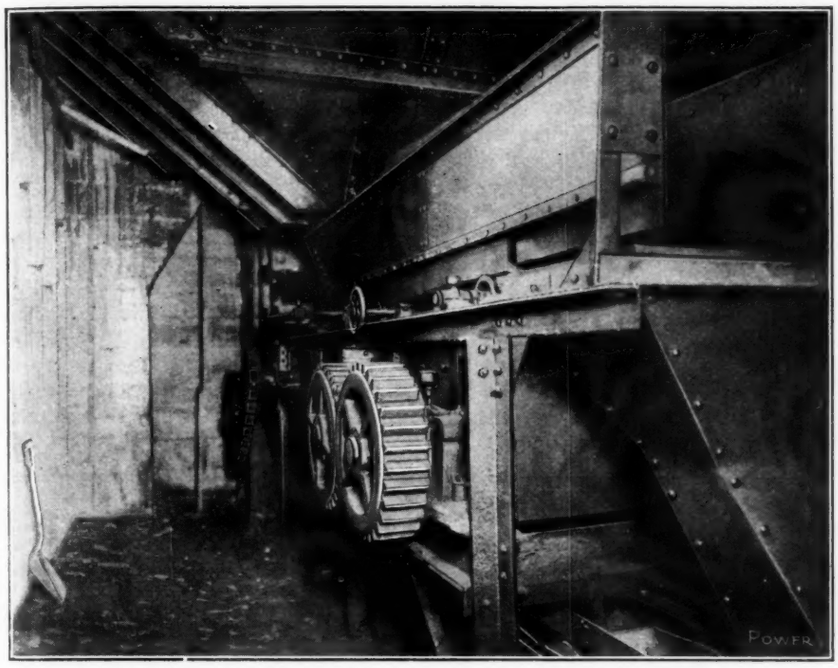


FIG. 4. FEEDER AND CRUSHER

end of the lower run of the conveyor, together with the take-ups and general type of construction, is shown in Fig. 2.

The equipment in the basement deserves special mention. This is seen in

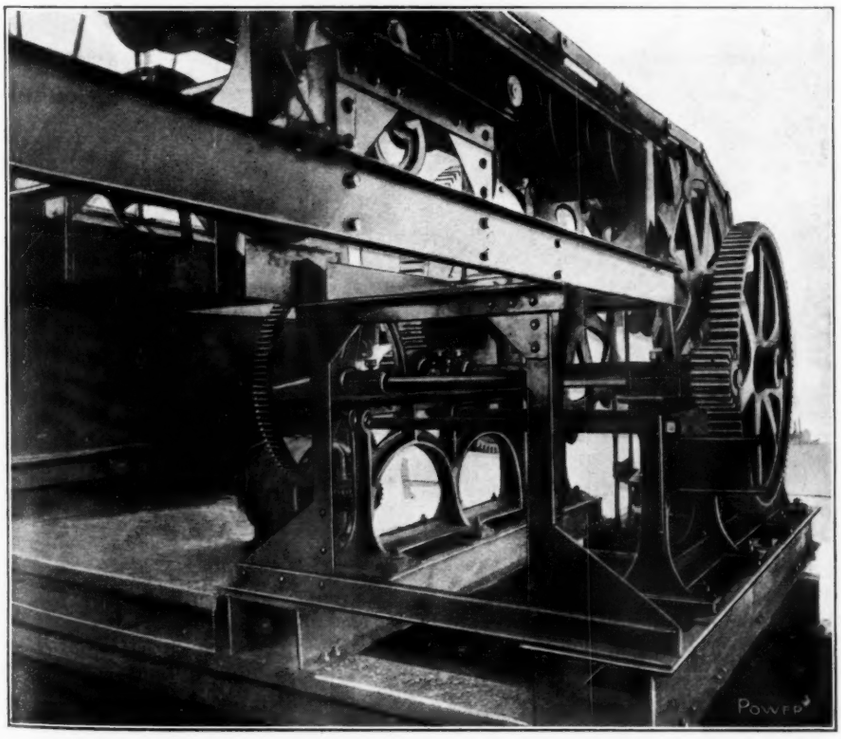


FIG. 5. DRIVE AT UPPER END OF ASCENDING STRAND

section *AA*, Fig. 1. The ash chutes traverse the full length of the space in front of the ashpits and fine-coal hoppers and may be placed as desired by the operator. The ashes are raked from the pits into these chutes and thence into the conveyor, no shoveling being required. These same chutes are also used to transfer to the conveyor coal dust from the fine-coal hoppers.

The capacity of the conveyor is 50 tons of coal per hour and 10 hp. is required for its operation. Considering the length of runs and the lift, this is low, but is due in great measure to the large self-oiling chain rollers which turn on extra-large case-hardened bushings, and because there is no scraping or dragging action, the material being carried throughout its entire course. The drive, which is placed at the upper end of the ascending strand, is shown in Fig. 5, and the opposite end of the upper run, with the tripper or dumper over the ash chute, is shown in Fig. 3.

Bracing Flat Surfaces in Steam Boilers

There is considerable variation as to the load allowed per square inch of net section on diagonal braces, rod braces and staybolts by the authorities who have laid down rules on this subject. The United States Government rules allow 6000 lb. on welded iron stays below 1¼ in., 7500 on 1¼ in. and above, and from 7000 to 9000 lb. on weldless steel stays. Chicago has a flat scale of 6000 lb. on all stays or braces, Philadelphia has a limit of 7500 lb.; the Massachusetts rules allow from 6500 to 9000 lb. per square inch net section, varying as the braces are welded or weldless and with the size, the latter for the reason that with a given waste of material the percentage of reduction is greater with the smaller sizes.

The above applies to flat surfaces and refers to flat heads, such as dome heads, segments of heads, etc. The United States Government has a rule to find the pressure on flat heads not exceeding 20 in. in diameter as follows:

$$P = \frac{C \times T^2}{A}$$

where

P = Pressure;
C = 112 ($\frac{7}{16}$ or under) and 120 over $\frac{7}{16}$);
A = ½ the area;
T = Thickness in sixteenths.

With a ¾-in. head 20 in. in diameter, 110 lb. would be allowed by these rules.

A short time ago the Board of Boiler Rules for the State of Ohio issued instructions to the inspectors holding certificates of competency that the following formula could be used in flat surface of heads:

$$P = \frac{T \times T.S. \times 10}{A \times 8}$$

where

T = Thickness;
$T.S.$ = Tensile strength;
A = Area;

In addition, a limitation allowance of 8000 lb. per square inch irrespective of size of brace is granted. This applies to boilers now in use, but not to boilers to be installed after July 1, 1912. This ruling is far more liberal than any other authority has heretofore allowed as a comparison will show. Assume a 72-in. boiler, height of segment 24 in., thickness of head ½ in., tensile strength 60,000 lb. The total area of the segment = 1186.4

sq.in., while the area requiring bracing = 814 sq.in. Hence

$$\frac{0.5 \times 60{,}000 \times 10}{814 \times 8} = 46\ lb.$$

allowed without braces.

Let the pressure required be equal to 100, then 100 — 46 = 54 lb. to be braced, and 54 × 814 = 43,956 lb. Assuming the proposed brace to be of 0.79 in. area, then 0.79 × 8000 = 6320 lb. per brace, and 43,956 ÷ 6320 = 7 braces of practically 1-in. diameter.

It may be said that flat surfaces subjected to internal pressure will spring, and proportionally to the unsupported area. Samuel Nichols, in his tests of circular flat heads, showed the springing began with very low pressures even at 20 lb. on 28-in. heads and increased as the pressure was raised. Applying this fact then to the Ohio ruling, it seems the head would so spring that at 100 lb. pressure the total load on the braces would be 0.79 × 7 = 5.53 into the total load, 81,400 lb., or 14,718 per square inch net section instead of 8000 lb. Applying this ruling to a flat dome head 36 in. diameter, ½ in. thick, 60,000 lb. tensile strength, area to be braced 707 in., gives 53 lb. without bracing. The results of allowing a flat head unbraced to spring and return times without number would be final failure due to such action.

Reverting to the segment as in a horizontal tubular boiler, it may be said other authorities have been careful to avoid allowing excessive stresses on the chord of the segment which is supported by the tubes inasmuch as the latter are not a constant in strength as is the flange of the head in the arc of the segment, and this view has been approved by most students as the tubes are subjected to more or less rapid wear and reduction in thickness.

Further, such calculations apply to boilers now in use irrespective of age. Indeed, Ohio has no limitation as to age as respects pressure to be determined with a factor of safety of 4 together with this exceedingly liberal allowance on braces. Comparing this with the Massachusetts, Chicago, Philadelphia and Detroit rules, what results may be expected?

Preparing for License Examination

By W. Nelson

This man worked out problems based on conditions existing in his power plant. He learned to "do things" by doing as well as by reading and finally won out.

Many engineers have criticised, justly and otherwise, license examiners and examinations, and occasionally a list of examination questions is submitted with the purpose of aiding others, but no one, to my knowledge, has proposed a plan of preparation for meeting the examiner.

My method was no short cut; it did not eliminate hard work, but it was so effective that every examination yielded the license sought.

Learn By Doing

I soon realized that one best learns by doing; that there is nothing like actually performing an operation to impress it upon the mind. Reading directions for babbitting crankpin brasses gives some ideas, but to make a lasting impression, start right in and clean, babbitt, bore and scrape them. When the job is completed, one knows how to do the work.

When I began studying boiler computations, I measured the shells and tubes of the boilers I was firing, calculated the heating surface, computed the efficiency of the joints and figured the allowable pressure. The coal slips were then added up and the coal burned per square foot of grate per hour calculated, and then I determined the amount of air necessary to burn that amount of coal.

The horsepower of every engine in the plant was computed and some in other plants. One engineer allowed me to take diagrams from his engine, loaning me his planimeter to measure the area.

I determined the cylinder ratios of the engine, the total expansion at several cutoffs, the size of condensers required and the amount of cooling water, the gain from heating the feed water, and the size of safety valves required; many other things I learned to do—by doing. When I learned how to work a problem, the idea was fixed in my mind by applying it to the apparatus about me.

Experimented on Old Engine

One day in my early training after reading how to put an engine on its centers, I was allowed to practice on an idle auxiliary engine. I made a tram, and after my watch was over I started my work. A month's reading could not have fixed the scheme in my mind as firmly as it was fixed in the next hour. The chief was interested, and had me put the engine on each center and mark each position on the guide. Then he took down the rod and showed me how to find the striking points and the clearance and how to assemble and key up the rod. In learning one thing well I also learned several others, and for keeps.

An acquaintance who had an old engine let me use it for practising valve setting, my watch engineer furnishing the instruction. After setting the valve several times, we took out the brasses and scraped them, keyed up and made a slight change in the clearance and several other adjustments. The engine was put in shape and I got some very practical instructions out of this day's work.

Formulas

To formulas I applied the principle of learning by doing. I selected some of the simplest and applied values from concrete examples about me. After mastering these, I gradually worked up to the more complex ones and soon found that my former fears were foolish, that the formula was a very useful servant. All that is necessary is a little patient work.

Many gain little from their reading because they shun the hard spots and the mathematics; they swallow things whole and accept statements without question. The man who challenges a statement until he proves it, conquers the tough passages and solves problems, gets something of value for the time spent.

Data Sheets

One chief gave me the idea of gathering data sheets of the plant apparatus. Upon his desk is a data-sheet folder. Each boiler, engine and device has a sheet. If he wants to order a repair part its dimensions are found on the card.

When I took charge of a plant I remembered these things and started a data book, taking what measurements I could with things running; the rest I got while the plant was shut down. The pumps and boilers received the same attention; in fact, everything of importance about the plant that I could think of was jotted down for the data sheets.

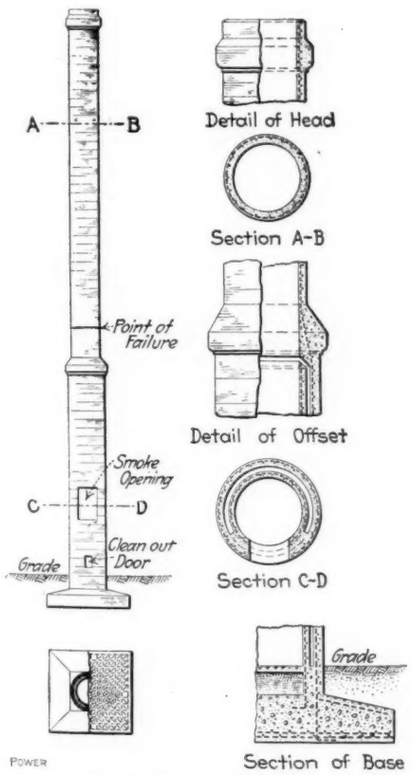
It meant work, but I was learning a lot that I otherwise would never have known. When I had opportunity I studied the data sheets. Working from the cylinder, I computed the required size of wristpin, crankpin, connecting-rod, etc., by several formulas. Next, using the shaft, pins and connecting-rod as a basis, I figured out how large a cylinder they would carry. The radiator surface required for the several rooms was computed by different methods and the results compared. The stress upon the boiler head above the tubes, the allowable pressure, and various other things about the boiler and other apparatus were computed. These sheets furnished a fine opportunity to learn by doing.

Another chief said: "Don't hurry or worry about a license. Try to fit yourself to be a real engineer. If you do that, the license will come." I took a course in a correspondence school which proved profitable and enjoyable. Engineering journals have been constant helps. One day I took up the questions in the questions and answers department, and before reading the answers I worked out ones of my own; then I compared them with those published.

To get a varied experience one should not work too long in one plant. Changing gives the engineer a broader view from contact with machinery and men. In a small plant, six months might suffice, but a year would be better. In a large plant, two years can be spent to good advantage. Visit plants in your vicinity and study them carefully.

Wind Wrecks Concrete Chimney

A cyclonic wind storm wrecked the 175-ft. reinforced-concrete chimney of the Pennsylvania Cement Co., Bath, Penn., at about 10 o'clock Sunday night, June 2, and caused the death of three men.



SHOWING DETAILS OF CONCRETE CHIMNEY CONSTRUCTION AND POINT OF FAILURE

The chimney was 8 ft. outside diameter from the lower double-shelf section to the top. The point of failure was 65 ft. from the base and the chimney was apparently sheared off at that height by the excessive wind pressure. The illustration shows the general design of the chimney and the point of failure.

From information obtainable at this writing it seems that the wind was exceptionally strong, sweeping a path approximately ¼ mile wide, uprooting trees, unroofing houses and also freight cars which were standing on a siding. From the appearance of the remaining portion of the stack and the fallen parts, the upper section, approximately over 100 tons, after breaking away, was blown sideways a distance of 30 ft. before it landed on the roof of the concrete boiler house, the striking point being indicated by a semicircular hole in the roof, showing that the fractured stack struck the roof an end blow, when the stack crumbled into small pieces. The concrete appeared to be porous and weak.

In breaking through the roof, a T-fitting on the main steam header was cracked and the 5-in. flanged saddle on one of the boilers was broken. The firemen remained at their post and immediately pulled the fires from under the nine 250-hp. boilers which the stack served.

They did not know of the death of their fellow workmen until after this work had been accomplished. It was then found that one fireman had been killed by some flying object and that two ash handlers had died from the effects of inhaled steam. None of the other firemen were injured and the property loss, outside of the chimney, will be small.

This chimney was built by a chimney construction company, now out of existence, and was finished in May, 1907. According to officials of the concrete company, there was trouble from the start regarding the methods of construction. Two rings of the stack above a height of 125 ft. were built during freezing weather. The work was stopped until spring and then, instead of removing the frozen rings, the remainder of the stack was built upon them. Because of this or some other cause the chimney was out of plumb 6 in. within 30 days after its completion. The chimney was plumb, however, for 125 ft. above the base.

This type of stack construction has been discarded by all concrete chimney builders both as to design and method of mixing the concrete material.

Recording Liquid Level Gage

There are several points of difference between this gage and that described in our issue of May 14. The pressure tube is so compact, being made with few diaphragms, that it can be located in the ordinary circular case, as shown in Fig. 1, and requires very little volume change and air movement to produce the indication. The movement of the diaphragm is restrained upon one side by one or more helical springs, two of which are shown at *SS* in the illustration, instead of by a flat metal strip, and the pen-arm is carried upon a shaft supported at both ends, the motion caused by the action of the pressure in the tube being transmitted to the shaft through a series of simple levers.

In the helical form of tube used for higher ranges the pen-arm is also fastened to the shaft, the movement being transmitted by a connection which is flexible radially. A friction joint is provided in the pen-arm to enable any necessary adjustment and this joint is in the line of the axis so that any change affects the adjustment of the pen on its proper arc without shortening the pen-arm or disturbing any other adjustments.

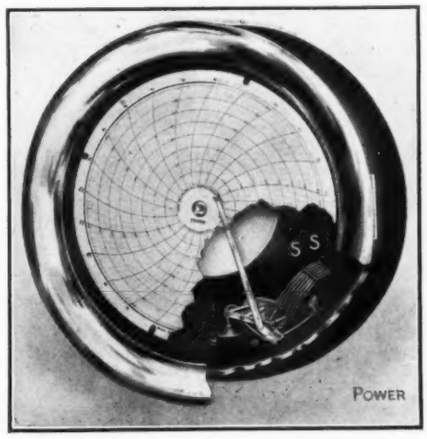


FIG. 1. RECORDING LIQUID LEVEL GAGE

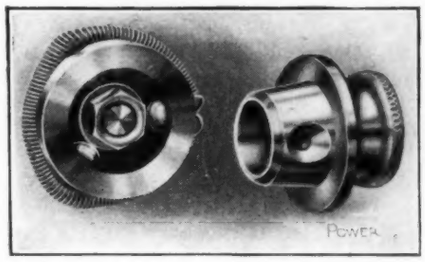


FIG. 2. SELF-CENTERING CHART HOLDER

The tension head of the chart holder, shown by Fig. 2, is tapered so as to automatically center the chart, and when removed leaves a flat surface on the dial, facilitating the putting on and taking off of the charts without disturbing the pen-arm or any other working part. The chart is held friction-tight between the upper and the lower surfaces of the hub and can be readily adjusted to the time position.

Skeleton and bold figures distinguish the day and night hours, as shown in Fig. 1. The clock movement has a 3-point support eliminating the possibility of strain from imperfect seating upon the case. All the cases are made dust and moisture proof. The Industrial Instrument Co., Foxboro, Mass., is the manufacturer.

Superheated Steam in Locomotive Service

In bulletin No. 57 issued by the University of Illinois Engineering Experiment Station, Prof. W. F. M. Goss reviews his previous work on the above subject carried on in the laboratory of Purdue University. The results of the study are given in the following paragraphs, and the bulletin giving the details of the tests may be obtained by addressing the author.

Foreign practice has proved that superheated steam may be successfully used in locomotive service without involving mechanism which is unduly complicated, or difficult to maintain.

There is ample evidence to prove that the various details in contact with the highly heated steam, such as the superheater, piping, valves, pistons and rod packing, as employed in German practice, give practically no trouble in maintenance; they are ordinarily not the things most in need of attention when a locomotive is held for repairs.

The results of tests confirm, in general terms, the statements of German engineers to the effect that superheating materially reduces the consumption of water and fuel and increases the power capacity of the locomotive.

The combined boiler and superheater tested contains 943 sq.ft. of water-heating surface and 193 sq.ft. of superheating surface; it delivers steam which is superheated approximately 150 deg. The amount of superheat diminishes when the boiler pressure is increased, and increases when the rate of evaporation is increased, the precise relation being

$$T = 123 - 0.265\,P + 7.28\,H$$

where

T = Superheat in degrees F.,
P = Boiler pressure by gage,
H = Equivalent evaporation per foot of water-heating surface per hour.

The evaporative efficiency of the combined boiler and superheater tested is

$$E = 11.706 - 0.214\,H$$

where E is the equivalent evaporation per pound of fuel and H is the equivalent evaporation per hour per foot of water-heating and superheating surface.

The addition of the superheater to a boiler originally designed for saturated steam involved some reduction in the total area of heat-transmitting surface, but the efficiency of the combination, when developing a given amount of power, was not lower than that of the original boiler.

The ratio of the heat absorbed per foot of superheating surface to that absorbed per foot of water-heating surface ranges from 0.34 to 0.53, the value increasing as the rate of evaporation is increased.

When the boiler and superheater are operated at normal maximum power, and when they are served with Pennsylvania or West Virginia coal of good quality, the available heat supplied is accounted for approximately in Table 1.

TABLE 1. AVAILABLE AND LOST HEAT

	Per Cent.
Absorbed by water	52
Absorbed by steam in superheater	5
Utilized	57
Lost in vaporizing moisture in coal	5
Lost in CO	1
Lost through high temperature of escaping gases	14
Lost in form of sparks and cinders	12
Lost through grate	4
Lost through radiation, leakage and unaccounted for	7

Water consumption under normal conditions of running has been established as in Table 2.

TABLE 2. WATER CONSUMPTION

Boiler Pressure, Lb.	Corresponding Steam per I.hp.-hr., Lb.
120	23.8
160	22.3
200	21.6
240	22.6

The minimum steam consumption for the several pressures is materially below the values given. The least for any test was 20.29 lb.

Table 3 gives the established coal consumption under normal conditions of running.

TABLE 3. COAL CONSUMPTION

Boiler Pressure, Lb.	Coal Consumed per I.hp.-hr., Lb.
120	3.31
160	3.08
200	2.97
240	3.12

Neither the steam nor the coal consumption is materially affected by considerable changes in boiler pressure, a fact which justifies the use of comparatively low pressures in connection with superheating.

Contrary to the usual conception, the conditions of cutoff attending maximum cylinder efficiency are substantially the same for steam superheated 150 deg. as for saturated steam. With superheated steam, when the boiler pressure is 120 lb., the best cutoff is approximately 50 per cent. stroke, but this value should be diminished as the pressure is raised, until at 240 lb., it becomes 20 per cent.

Tests under low steam pressures, for which the cutoff is later than half stroke, give evidence of superheat in the exhaust. The saving in water consumption and in coal consumption per unit of power developed, which was effected by the superheating locomotive, Schenectady No. 3, in comparison with the saturated-steam locomotive, Schenectady No. 2, is given in Table 4.

The power capacity of the superheating locomotive is greater than that of the saturated-steam locomotive.

TABLE 4. COMPARATIVE ECONOMY OF LOCOMOTIVES USING SATURATED AND SUPERHEATED STEAM

Saving in Water Consumption				Saving in Coal Consumption			
	Locomotive				Locomotive		
Boiler Pressure, Lb.	Saturated Steam, Lb.	Superheating, Lb.	Gain, Per Cent.	Boiler Pressure, Lb.	Saturated Steam, Lb.	Superheating, Lb.	Gain, Per Cent.
120	29.1	23.8	18	120	4.00	3.31	17
160	26.6	22.3	16	160	3.59	3.08	14
200	25.5	21.6	15	200	3.43	2.97	13
240	24.7	22.6	9	240	3.31	3.12	6

Condenser Performance and Economical Vacuum

The report of the Prime Movers Committee of the National Electric Light Association treats the above subject as follows:

Radical changes have been made during the past two years in condenser design, the effort being to so arrange the tubes and baffle the flow of steam that the steam side of the tubes will be readily freed of condensation and that the entrained air will be swept toward the dry vacuum pump suction and over a nest of special tubes designed to condense any remaining steam in the air, thus reducing the volume of the vapor handled by the pump; but probably the greatest improvement has come about through a realization that additional cooling surface in a condenser is of little value unless ample provision is made for the steam to reach it without undue loss of pressure.

The accompanying curve shows how the theoretical thermal efficiency of a perfect steam engine varies with terminal pressures and indicates that the most promising field of endeavor for increased efficiency in steam-turbine work lies in the production and maintenance of the highest vacuums obtainable. The reduction of terminal pressure from 0.4 lb. to 0.2 lb. indicates an addition of about 8 per cent. in the theoretical amount of energy available, and this saving is double the amount which would be saved by a reduction in pressure from 0.6 lb. to 0.4 lb.

It has often been demonstrated that the vacuum may be improved by removing many of the tubes to provide liberal passages for the steam to distribute among the remaining tubes without excessive velocities or loss of pressure.

Another advantage to be obtained by the opening of steam passages and a reduction of steam velocities is the elimination of the whirling and creeping of tubes, which result in their becoming loose in the packing and even in being worn through in the tube plate.

The importance of this is apparent when it is considered that a 12,000-kw. turbine, when running at full load and with a condenser pressure of ½ lb. absolute pressure, is discharging into the condenser each second 6000 cu.ft. of steam; and if in the first pass of the condenser there is between all of the tubes of the first row only 10 sq.ft. of passage, the steam must have a velocity of 600 ft. per second, which can only be attained at the expense of a considerable amount of back pressure on the turbine, with a corresponding loss of efficiency.

Where condensing water is plentiful, but so dirty that the amount flowing through the condenser is restricted, due to the tubes being choked, relief may be obtained by the installation of grid-iron valves so located that the path of the circulating water may be readily changed from 3-pass to single-pass, thus reducing the friction and increasing the amount of cooling water.

Life of Condenser Tubes

There has been very little development during the past year on this subject. One large company reports that it is experimenting with tubes of different compositions, the two principal types being Admiralty (70 per cent. copper, 29 per cent. zinc and 1 per cent. tin) and Benedict bronze (92 per cent. copper and 8 per cent. aluminum). Sufficient time, however, has not elapsed to warrant the drawing of any conclusions.

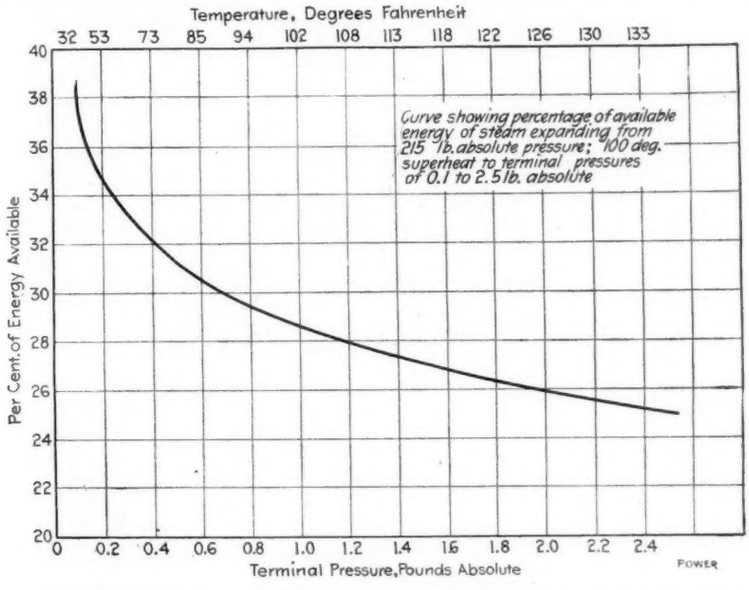
Another company that has had very serious tube troubles reports an installation of Benedict bronze tubes in a 32,000-ft. condenser for nearly three years with no indication of deterioration of any kind, while a similar installation a few months old is quite disappointing.

The so called Cumberland protective process is being tried out in several stations. The method employed consists of passing a small current of from 10 to 20 amperes through the tubes of the condenser to the ground in such a way that the tubes are made electro-negative to the water surrounding them. This system has not been in service a sufficient time to warrant any statements in regard to its value.

Combustion of Coal*

By E. A. Uehling

Every pound of coal of a given composition contains a definite amount of potential energy. Combustion liberates this energy in the form of heat energy. To produce combustion two factors are necessary: oxygen and high temperature.


Temperature, Degrees Fahrenheit
32 53 73 85 94 102 108 113 118 122 126 130 133
Curve showing percentage of available energy of steam expanding from 215 lb. absolute pressure; 100 deg. superheat to terminal pressures of 0.1 to 2.5 lb. absolute
Per Cent. of Energy Available
40 38 36 34 32 30 28 26 24 22 20
0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0 2.2 2.4
Terminal Pressure, Pounds Absolute
POWER

Combustion in the sense under consideration is the chemical reaction between oxygen (O) and carbon (C) forming carbon dioxide (CO_2), and between hydrogen (H) and oxygen forming water vapor; that is, steam (H_2O).

This chemical reaction, called combustion, liberates the potential energy in coal in the form of heat energy. For every unit of carbon and every unit of hydrogen, a fixed quantity of heat energy is liberated. This heat energy becomes effective at once by heating the solid carbon and ash of the coal to incandescence and raising the gaseous products of combustion to a high or flame temperature. The highest flame temperature results from complete combustion with the theoretically correct quantity of oxygen. In practice theoretically perfect combustion is impossible, but complete combustion is attainable, and, to produce the greatest useful heat effect, should be accomplished with a minimum excess of oxygen.

The rapidity of combustion depends on the temperature of the fire, and the quantity of oxygen supplied.

The oxygen required for the combustion is derived from the atmosphere, which is composed of a mixture of 23 parts oxygen and 77 parts nitrogen by weight. Nitrogen retards combustion; hence too great an excess of oxygen with its accompanying fourfold quantity of nitrogen will tend to bring the flame temperature below the point of ignition.

In the burning of coal, decomposition always precedes combustion; hence the greater the amount of volatile combustible contained in a coal the greater the flame volume, and the greater the length of flame the greater the loss of heat from radiation and the probability of incomplete combustion because of low-flame temperature.

Smoke is the usual manifestation of incomplete combustion. It results from three causes: low-flame temperature, stratification of air and the volatile combustible gases, and deficiency of oxygen. Stratification is caused by incompetent or careless firing, leaving an uneven fuel bed. The air following the course of least resistance rushes through the light covering in excess of the requirement and in inadequate quantity through the heavier portions, resulting in strata with oxygen in great excess and strata with deficiency of oxygen, all below the temperature which would result from proper intermingling of the air with the combustible gases, the one because of a deficiency in combustible and the other because of a lack of oxygen, all quickly becoming cooled below the point of ignition by giving up their heat to the boiler before opportunity for combustion was offered by proper intermingling.

That decomposition precedes combustion is not only true of coal, but also for the hydrocarbons. Heat separates the heavy hydrocarbons into free carbon and light hydrocarbons, and before combustion can take place the latter are further separated into carbon and hydrogen. In the act of combustion two atoms of hydrogen combine with one atom of oxygen, forming H_2O, water vapor, and one atom of carbon combines with two atoms of oxygen, forming CO_2, carbon dioxide.

The combustion of the hydrogen precedes that of the carbon and liberates an intense heat which raises the temperature of the carbon to incandescence, producing a luminous flame, which, if brought in contact with a cold body, produces smoke and soot. The same takes place if the flame temperature is reduced from any other cause, such as rapid radiation or admixture of an excess of

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cold air. Too great an excess of air is a prolific source of incomplete combustion.

To secure complete combustion there must be had a hot furnace, a moderate excess of air and a thorough intermingling of the air with the volatile combustibles. The first means a furnace so constructed that combustion will be complete before the temperature of the flame is materially reduced; the second, that the air supply must be controlled so as to avoid too great an excess; and the third, that the firing must be judiciously and skillfully done and the air admitted in the proper manner.

The object of burning coal under a steam boiler is to convert the chemical potential energy contained in the coal into mechanical potential energy, in the form of steam under pressure. Heat is the medium of transfer and the problem before the combustion engineer is, first, how to burn the coal completely, and second, how to get the maximum amount of heat liberated by combustion into the water and steam contained in the boiler under which the coal is burned.

The greatest single loss is the heat carried away by the flue gases. This increases as the difference in temperature between the flame and the flue gases decreases, and increases directly as the weight and temperature of these gases increase; now since the weight of the flue gas per unit of combustible increases directly with the excess of air supplied, and since the temperature of the flame is decreased thereby and that of the flue gas is increased, the importance of regulating the air supply becomes evident.

It is practically impossible to attain complete combustion of coal with the theoretical amount of air required to oxidize fully all its combustible elements. The amount of air required for maximum economy will vary with the chemical nature and the physical condition of the coal, with the type and construction of furnace and boiler, the availability and regulation of the draft and the manner of stoking; but for any one of these or combination of these conditions, there is a minimum excess of air that will produce maximum economy. Every pound of air in excess of that required carries heat to waste and a very great excess may even be the cause of incomplete combustion, thus proving wasteful at both ends.

Since every pound of combustible contained in a given coal carries a definite per cent. of carbon, it follows that in complete combustion with the theoretically required amount of air, the products of combustion must contain a definite per cent. of CO_2. It further follows that the per cent. of CO_2 must decrease as the excess of air increases and that CO_2 is, therefore, a correct index of the loss of sensible heat up the chimney.

Sporadic determinations of CO_2 are of small value, and unless judiciously taken, may even be quite misleading. The CO_2 from an average sample collected uniformly over a day's run is more valuable, since it shows the average efficiency of the fire, and in connection with the factor of evaporation will be of great aid in determining the conditions for improving economy; but here again the results may be misleading, especially in connection with hand-fired boilers. A proper per cent. of CO_2 in an average sample spread over a day's run is no guarantee of regular and efficient stoking. At some intervals the CO_2 may have been too high and at others too low for efficient work, causing a loss of combustibles due to deficiency of oxygen during the former, and a loss of sensible heat due to too great an excess of air during the latter intervals. These uncertainties are eliminated by employing an apparatus which will continuously indicate and autograhpically record the per cent. of CO_2 contained in the flue gases. Such a record is not only of the greatest assistance in determining the excess of air required for maximum efficiency, but it is necessary also to be able to maintain the high efficiency which it has been possible to attain by intelligent application of theory and practice to the existing conditions.

Sawmill Boiler Explosion

By W. J. Smith

A disastrous boiler explosion occurred at the Orillia Lumber Co. mill, about two miles west of Orillia, Wash., on the morning of Apr. 11. Three men were injured, one of whom died later.

The boiler and engine house and a considerable portion of the mill structure were destroyed, and the loss was several thousand dollars.

The boiler was about five years old, of the horizontal tubular type, 66 in. in diameter by 16 ft. long, made up in three courses; it had double-riveted lap seams. The boiler was designed to operate at about 110 lb. and this seems to have been the pressure carried. The failure occurred at the longitudinal seam of the rear course and was caused by a lap seam crack underneath the edge of the inner plate which gave no evidence of its presence until complete rupture occurred.

Subsequent examination revealed that the fracture, which was blackened and dull, must have existed for some time. It extended about 4 in. from the head seam to the same distance from the girth seam, and nearly severed the plate, some $\frac{1}{32}$ to $\frac{1}{16}$ in. of metal failing when the explosion occurred. What is left of the boiler is shown in the illustration.

The rear course apparently whipped off, snapping the rivets from both head and girth seams. When near the bottom a rending or tearing of the second course occurred and extended longitudinally into and through the first course. This metal showed a reduction of thickness due to the terrific strain imposed before these parts failed.

The braces supporting the heads above the tubes were of the round, upset or weldless type. One of the interesting features is the demonstrated high efficiency or holding power of this form of brace; though bent nearly at right angles to the shank or palm, only one was broken. The feet were pulled from a T- to a Y-form. In many instances the rivets fastening these ends and being in tension were reduced in area nearly 50 per cent. before giving way.

Close examination of the plate which failed first revealed that it was defective, due possibly to some constituent impurity. The openings for the safety valve and blowoff were both in this sheet and radiating from the punched holes on each opening were several small cracks. As these parts had never been under a strain, they indicate a brittle, unreliable material.

The rear course, opening on the sides and near the top, was driven downward and remained near the point of the accident, but the first and second courses, opening at or near the bottom, were lifted from their position and at the same time driven forward, bending the front head at right angles, stripping it from the tubes and then crashing through the roof of the building.

Remains of Exploded Sawmill Boiler

Horsepower and Kilowatts

By F. R. Low

A horsepower is equal to 745.65 watts or 0.74565 of a kilowatt.

A kilowatt is equal to 1000 watts or

$$1000 \div 0.74565 = 1.3411 \; hp.$$

These factors will do for the abstract or mathematical conversion of horsepower and kilowatts, but in their physical conversion there is always a loss. If the efficiency of a unit is 90 per cent.—that is, if the electric energy at the terminals of the generator is only 90 per cent. of the mechanical energy applied to its production—one horsepower so applied will produce only $0.74565 \times 0.90 = 0.67108$ of a kilowatt and it will take

$$\frac{1}{0.74565 \times 0.90} = 1.49012 \; hp.$$

to produce a kilowatt.

For each pound of steam required per kilowatt at 90 per cent. efficiency there would be required 0.67108 lb. per horsepower; or for each pound required per horsepower 1.49012 lb. per kilowatt.

The accompanying table gives these factors for efficiencies from 67 per cent., where it takes practically 2 hp. to make a kilowatt, to 100 per cent., advancing by tenths of a per cent. The factors for the even percentages are given in the second column under zero, and those for the successive tenths under the columns so respectively denominated.

For example, with an efficiency of 88.5 per cent. it will be found in the line opposite 88 and under 0.5 that 1 hp. will produce 0.6599 of a kilowatt and that it will take 1.51538 hp. to produce a whole kilowatt.

A few examples will make the application and use of the table apparent.

EQUIVALENT HORSEPOWER AND KILOWATTS AT VARIOUS EFFICIENCIES

Efficiency, Per Cent.	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	Efficiency, Per Cent.
99	0.73819 1.35466	0.73894 1.35329	0.73968 1.35193	0.74043 1.35057	0.74118 1.34921	0.74192 1.34785	0.74267 1.34650	0.74341 1.34515	0.74416 1.34380	0.74490 1.34245	99
98	0.73074 1.36848	0.73148 1.36709	0.73223 1.36569	0.73297 1.36430	0.73372 1.36292	0.73447 1.36153	0.73521 1.36015	0.73596 1.35878	0.73670 1.35740	0.73745 1.35602	98
97	0.72328 1.38259	0.72403 1.38117	0.72477 1.37974	0.72552 1.37833	0.72626 1.37691	0.72701 1.37550	0.72775 1.37409	0.72850 1.37268	0.72925 1.37128	0.72999 1.36988	97
96	0.71582 1.39699	0.71657 1.39554	0.71732 1.39409	0.71806 1.39264	0.71881 1.39119	0.71955 1.38981	0.72030 1.38831	0.72104 1.38688	0.72179 1.38545	0.72253 1.38402	96
95	0.70837 1.41170	0.70911 1.41021	0.70986 1.40873	0.71060 1.40725	0.71135 1.40578	0.71210 1.40431	0.71284 1.40284	0.71359 1.40137	0.71433 1.39991	0.71508 1.39845	95
94	0.70091 1.42671	0.70166 1.42520	0.70240 1.42369	0.70315 1.42218	0.70389 1.42067	0.70464 1.41917	0.70538 1.41767	0.70613 1.41617	0.70688 1.41467	0.70762 1.41318	94
93	0.69345 1.44206	0.69420 1.44051	0.69495 1.43896	0.69569 1.43742	0.69644 1.43588	0.69718 1.43434	0.69793 1.43281	0.69867 1.43128	0.69942 1.42976	0.70017 1.42823	93
92	0.68600 1.45773	0.68674 1.45615	0.68749 1.45457	0.68823 1.45299	0.68898 1.45142	0.68973 1.44985	0.69047 1.44828	0.69122 1.44660	0.69196 1.44516	0.69271 1.44361	92
91	0.67854 1.47375	0.67929 1.47213	0.68003 1.47052	0.68078 1.46891	0.68152 1.46730	0.68227 1.46570	0.68301 1.46410	0.68376 1.46250	0.68451 1.46091	0.68525 1.45932	91
90	0.67109 1.49012	0.67183 1.48847	0.67258 1.48682	0.67332 1.48517	0.67407 1.48353	0.67481 1.48189	0.67556 1.48026	0.67630 1.47862	0.67705 1.47700	0.67780 1.47537	90
89	0.66363 1.50687	0.66437 1.50518	0.66512 1.50349	0.66587 1.50180	0.66661 1.50012	0.66736 1.49845	0.66810 1.49678	0.66885 1.49512	0.66959 1.49344	0.67034 1.49178	89
88	0.65617 1.52400	0.65692 1.52226	0.65766 1.52053	0.65841 1.51881	0.65915 1.51709	0.65990 1.51538	0.66065 1.51367	0.66139 1.51196	0.66214 1.51026	0.66288 1.50856	88
87	0.64872 1.54151	0.64946 1.53974	0.65021 1.53798	0.65095 1.53621	0.65170 1.53445	0.65244 1.53270	0.65319 1.53095	0.65394 1.52920	0.65468 1.52746	0.65543 1.52572	87
86	0.64126 1.55943	0.64020 1.55714	0.64275 1.55581	0.64350 1.55401	0.64424 1.55221	0.64499 1.55042	0.64573 1.54863	0.64648 1.54684	0.64722 1.54506	0.64797 1.54328	86
85	0.63380 1.57778	0.63455 1.57592	0.63529 1.57407	0.63604 1.57223	0.63679 1.57039	0.63753 1.56855	0.63828 1.56672	0.63902 1.56489	0.63977 1.56307	0.64051 1.56125	85
84	0.62635 1.59655	0.62709 1.59466	0.62784 1.59277	0.62858 1.59088	0.62933 1.58900	0.63007 1.58711	0.63082 1.58524	0.63157 1.58337	0.63231 1.58150	0.63306 1.57964	84
83	0.61889 1.61580	0.61964 1.61385	0.62038 1.61191	0.62113 1.60998	0.62187 1.60805	0.62262 1.60612	0.62336 1.60420	0.62411 1.60228	0.62485 1.60037	0.62560 1.59846	83
82	0.61143 1.63550	0.61218 1.63351	0.61292 1.63152	0.61367 1.62954	0.61442 1.62756	0.61516 1.62559	0.61591 1.62362	0.61665 1.62166	0.61740 1.61970	0.61814 1.61775	82
81	0.60398 1.65569	0.60472 1.65365	0.60547 1.65162	0.60621 1.64958	0.60696 1.64756	0.60770 1.64554	0.60845 1.64352	0.60920 1.64151	0.60994 1.63950	0.61069 1.63750	81
80	0.59652 1.67639	0.59727 1.67430	0.59801 1.67221	0.59876 1.67013	0.59950 1.66805	0.60025 1.66598	0.60099 1.66391	0.60174 1.66185	0.60249 1.65979	0.60323 1.65774	80
79	0.58906 1.69761	0.58981 1.69546	0.59055 1.69332	0.59130 1.69119	0.59205 1.68906	0.59279 1.68693	0.59354 1.68481	0.59428 1.68270	0.59503 1.68059	0.59577 1.67849	79
78	0.58161 1.71937	0.58235 1.71717	0.58310 1.71498	0.58384 1.71279	0.58459 1.71060	0.58534 1.70842	0.58608 1.70625	0.58683 1.70408	0.58757 1.70192	0.58832 1.69976	78
77	0.57415 1.74170	0.57490 1.73944	0.57564 1.73719	0.57639 1.73494	0.57713 1.73270	0.57788 1.73047	0.57862 1.72824	0.57937 1.72601	0.58012 1.72379	0.58086 1.72158	77
76	0.56669 1.76462	0.56744 1.76230	0.56819 1.75999	0.56893 1.75768	0.56968 1.75538	0.57042 1.75309	0.57117 1.75080	0.57191 1.74852	0.57266 1.74624	0.57340 1.74397	76
75	0.55924 1.78815	0.55998 1.78577	0.56073 1.78339	0.56147 1.78102	0.56222 1.77866	0.56297 1.77631	0.56371 1.77396	0.56446 1.77161	0.56520 1.76928	0.56595 1.76695	75
74	0.55178 1.81231	0.55253 1.80987	0.55327 1.80743	0.55402 1.80500	0.55476 1.80257	0.55551 1.80015	0.55625 1.79774	0.55700 1.79533	0.55775 1.79293	0.55849 1.79054	74
73	0.54432 1.83714	0.54507 1.83463	0.54582 1.83212	0.54656 1.82962	0.54731 1.82713	0.54805 1.82464	0.54880 1.82216	0.54954 1.81969	0.55029 1.81722	0.55104 1.81477	73
72	0.53687 1.86266	0.53761 1.86007	0.53836 1.85750	0.53910 1.85493	0.53985 1.85236	0.54060 1.84981	0.54134 1.84726	0.54209 1.84472	0.54283 1.84219	0.54358 1.83966	72
71	0.52941 1.88889	0.53016 1.88623	0.53090 1.88358	0.53165 1.88094	0.53239 1.87831	0.53314 1.87568	0.53389 1.87306	0.53463 1.87045	0.53538 1.86784	0.53612 1.86525	71
70	0.52196 1.91587	0.52270 1.91314	0.52345 1.91042	0.52419 1.90770	0.52494 1.90499	0.52568 1.90229	0.52643 1.89959	0.52717 1.89690	0.52792 1.89423	0.52867 1.89155	70
69	0.51450 1.94364	0.51524 1.94083	0.51599 1.93802	0.51674 1.93523	0.51748 1.93244	0.51823 1.92966	0.51897 1.92688	0.51972 1.92412	0.52046 1.92136	0.52121 1.91861	69
68	0.50704 1.97222	0.50779 1.96933	0.50853 1.96644	0.50928 1.96356	0.51002 1.96069	0.51077 1.95783	0.51152 1.95497	0.51226 1.95213	0.51301 1.94929	0.51375 1.94646	68
67	0.49959 2.00166	0.50033 1.99868	0.50108 1.99570	0.50182 1.99274	0.50257 1.98978	0.50331 1.98683	0.50406 1.98389	0.50481 1.98096	0.50555 1.97804	0.50630 1.97513	67

How many kilowatts would a brake load of 896.45 hp. produce when applied to a generator having an efficiency of 91.6 per cent.?

In the line opposite 91 per cent. and in the column under 0.6 it is shown that with an efficiency of 91.6 per cent. 1 hp. will develop 0.68301 kw. The given horsepower will therefore produce

$$896.45 \times 0.68301 = 612.28 \text{ kw.}$$

What indicated horsepower would have to be developed in an engine which was part of a unit having an overall efficiency of 87.3 per cent. to produce 1200 kw.?

With an efficiency of 87.3 the table shows that it takes 1.53621 hp. to produce a kilowatt. To produce 1200 kw. would therefore require

$$1200 \times 1.53621 = 1843.45 \text{ hp.}$$

An engine has a mechanical efficiency of 95 per cent. and the generator to which it is attached an efficiency of 94 per cent. What would be the indicated horsepower when the unit was turning out 2500 kw.?

The combined or overall efficiency of the engine and generator would be

$$95 \times 94 = 89.3 \text{ per cent.}$$

The table shows that at this efficiency it takes 1.5018 hp. to produce a kilowatt. To produce 2500 kw. would therefore require

$$2500 \times 1.5018 = 3754.5 \text{ hp.}$$

What would be the brake or shaft horsepower for the same output?

The generator converts the shaft energy with an efficiency of 94 per cent., at which the table says it takes 1.42671 hp. to make a kilowatt. To make 2500 kw. would therefore require

$$2500 \times 1.42671 = 3566.77 \text{ b.hp.}$$

This is, as it should be, 95 per cent. of the indicated horsepower

$$3754.5 \times 0.95 = 3566.77$$

A turbine-driven unit with an overall efficiency of 90.6 per cent. requires 16.5 lb. of steam per kilowatt-hour; what is the water rate per indicated horsepower?

At 90.6 per cent. the table shows that the kilowatts are 0.67556 of the horsepower. The steam consumption per horsepower is therefore 0.67556 times that per kilowatt or for the present case

$$0.67556 \times 16.5 = 11.1467 \text{ lb. per i.hp.}$$

There need be no uncertainty which of the two numbers in the table to use. It is evident that the steam per horsepower will be less than that per kilowatt, and that the smaller number is to be used as a multiplier in this case, while if the water rate is given in pounds per horsepower-hour it should be multiplied by the larger value in the table to obtain the water rate per kilowatt-hour.

The Quality of Steam in Boilers

By O. Z. Howard

How to make a calorimeter from pieces of pipe and standard fittings; also the method of figuring the quality of the steam and making corresponding corrections in the total evaporation.

An essential feature of a boiler test that is frequently omitted is the determination of moisture in the steam. While this may be so small as to affect the result but little in some cases, in others it may affect the equivalent evaporation as much as 2 or 3 per cent.

If a calorimeter is not available, one can be made very easily from pipe and standard figures like the illustration. The perforated nipple, with the long thread, should be screwed into a vertical portion of the steam pipe, as near the boiler as possible. The nipple should be of such length that the ⅛-in. holes near the ends will be about ½ in. from the sides of the steam pipe when screwed in place. The calorimeter should be thoroughly lagged with hair felt, from the point where it leaves the steam pipe to the elbow below the lower thermometer well.

The thermometer wells should be filled with mercury or a heavy oil, and a second thermometer should be suspended, in the open air, near each of those in the wells, to obtain the temperature of their stems, for making corrections as shown later. A manometer gage should be mounted on the side of the lower tee for measuring the back pressure in the exhaust pipe. An ordinary U-tube, as commonly used for taking boiler draft, will answer all requirements if filled with mercury in place of water.

If the amount of evaporation is determined by weighing the water before entering the boiler, no account of the loss of steam from the calorimeter will be necessary; but if the steam from the boiler is condensed and weighed, a correction for this loss will have to be made. This can be accomplished by running the exhaust into a barrel partly filled with water, thus condensing the steam and then weighing; by calibrating the orifice for the pressure used during the test, or by calculating the discharge. This latter method is correct enough for all practical purposes, and the discharge in pounds per minute, for an orifice ⅛ in. in diameter, will be $0.0105P$, where P is the absolute boiler pressure.

Before readings are taken on the calorimeter, the ½-in. valve should be opened wide for 10 or 15 min., then frequent readings of the four thermometers and the gage should be noted. The reading of the thermometer in the upper well, if properly corrected, should be a check on the gage reading. Assuming that there is no loss of heat by radiation in passing through the instrument, the total heat in the mixture of steam and water before throttling and in the superheated steam after throttling, will be the same, and can be expressed by the equation

$$xr + q = H_c + 0.476\,(T_1 - T_c)$$

or

$$x = \frac{H_c - q + 0.476\,(T_1 - T_c)}{r} \qquad (1)$$

where

x = Quality of the steam;
r = Latent heat of the steam at the pressure in the main;
q = Heat of liquid due to the pressure in the main;
H_c = Total heat in 1 lb. of dry steam at the calorimeter pressure (shown by manometer gage);
T_1 = Temperature in calorimeter (shown by lower thermometer);
T_c = Normal temperature of steam in calorimeter due to calorimeter pressure.

Thermometer Corrections

In general, all thermometers are calibrated for total immersion; that is, for the condition where both the stem and the bulb are at the same temperature. But in taking the temperature of the steam in a calorimeter some of the stem projects into space colder than the bulb. That being the case, a correction must be applied to the observed reading in addition to the correction found by the previous calibration of the thermometer.

The Bureau of Standards gives the following formula:

$$\text{Stem correction} = 0.000092 \times n\,(T - t) \qquad (2)$$

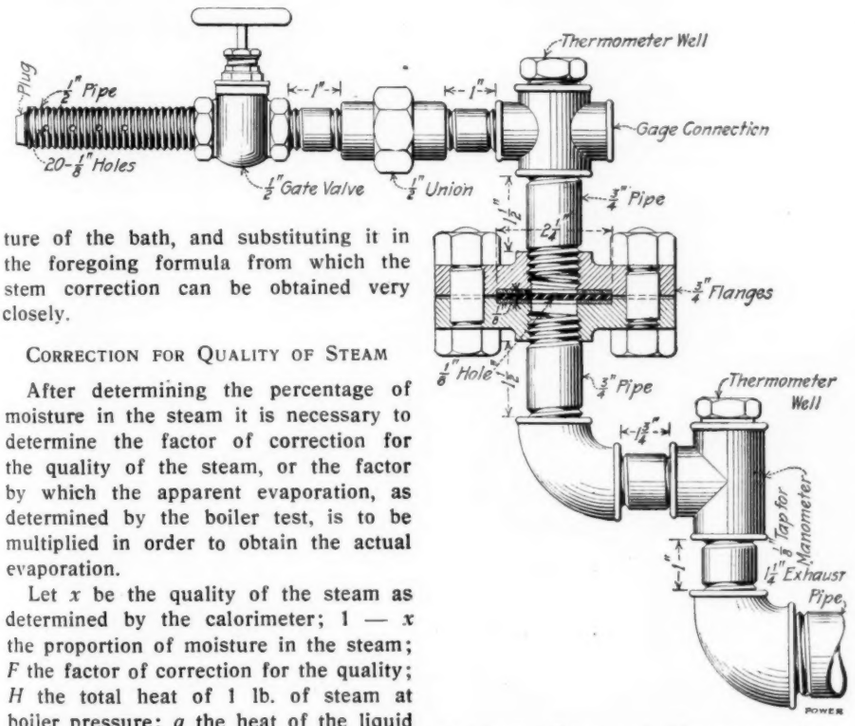
where

n = Number of degrees projecting from bath;
T = Temperature of the bath;
t = Mean temperature of the projecting stem.

The mean temperature t may be measused approximately by means of an auxiliary thermometer suspended near-by.

If the thermometer is used under conditions such that it has only a small stem correction, the reading of the ther-

mometer corrected by the mean of the table (from previous calibration) may be taken as the temperature of the bath, since a difference in the bath temperature makes a much smaller difference in the value of the stem correction. If, however, the number of graduations projecting from the well and the difference of temperature between the bath and the space above it are large, it will be necessary to determine the stem correction approximately and apply this to the corrected reading of the thermometer in order to get the approximate temperature of the bath. This may be done by using the reading of the thermometer, corrected from the table, as the temperature of the bath, and substituting it in the foregoing formula from which the stem correction can be obtained very closely.



SAMPLING TUBE AND HOMEMADE CALORIMETER

CORRECTION FOR QUALITY OF STEAM

After determining the percentage of moisture in the steam it is necessary to determine the factor of correction for the quality of the steam, or the factor by which the apparent evaporation, as determined by the boiler test, is to be multiplied in order to obtain the actual evaporation.

Let x be the quality of the steam as determined by the calorimeter; $1 - x$ the proportion of moisture in the steam; F the factor of correction for the quality; H the total heat of 1 lb. of steam at boiler pressure; q the heat of the liquid at the temperature due to boiler pressure, and h the heat of the liquid at the feed-water temperature.

The proportional part x receives from the boiler the sensible and the latent heat, or, the total heat in this portion above the temperature of the feed water is $x\ (H - h)$ B.t.u.; and the part $(1 - x)$ representing the water in 1 lb. of the mixture is $(1 - x)\ (q - h)$ B.t.u. But if all the water had been evaporated, each pound would have received the total heat in the steam above the temperature of the feed water, or $(H - h)$ B.t.u.

Hence,

$$F = \frac{x\,(H - h) + (1 - x)\,(q - h)}{(H - h)}$$

or

$$F = x + \frac{(1 - x)(q - h)}{(H - h)} \qquad (3)$$

Example: From a boiler test the following data are obtained with a throttling calorimeter: Steam gage, 144 lb. per square inch; temperature in the upper well, when immersed to the 200-deg. mark, 361 deg. F.; auxiliary thermometer at the upper well, 102 deg. F.; temperature in the lower well, 260 deg., when immersed to 240 deg.; auxiliary thermometer at the lower well, 102 deg.; manometer gage, 0.42 in. of mercury (corresponding to a pressure of 0.21 lb. per square inch), and the barometer (corrected) stood at 29.77 in.

Assume that from the calibration of the thermometers before the test, the reading of the thermometer in the upper well at 361 deg. was 2 deg. too high, and the reading of the thermometer in the lower well at 260 deg. was 1 deg. too low. Then from equation (2) the stem correction for the upper thermometer is

$$0.000092 \times 161\ (359 - 102) = 3.8\ deg.$$

and the true reading will be

$$361 + 3.8 - 2 = 362.8\ deg.\ F.$$

As so little of the stem of the lower thermometer projects, the stem correction need not be taken into account, and the true reading will be

$$260 + 1 = 261\ deg.\ F.$$

The reading of the barometer, 29.77 in., corresponds to a pressure of 14.62 lb. per square inch; hence the absolute steam pressure is

$$144 + 14.62 = 158.62\ lb.$$

per square inch, which from the steam table corresponds to a temperature of 362.9 deg., agreeing within 0.1 deg. of the temperature as obtained by the thermometer.

The quality of the steam may now be obtained. From the steam tables: r for saturated steam at 362.8 deg. F. equals 859.6 B.t.u.; q for saturated steam at 362.8 deg. F. equals 334.8 B.t.u.; H_c for 1 lb. of dry steam at 14.83 lb. (14.62 + 0.21) pressure absolute equals 1150.5 B.t.u.; T_1 from the corrected temperature in the lower well equals 261 deg. F., and T_c for the steam at 14.83 lb. pressure equals 212.41 deg. F.

Substituting in formula (1),

$$x = [1150.5 - 334.8 + 0.476\ (261 - 212.41)] \div 859.6 = 0.9758$$

or, the steam is 97.58 per cent. dry.

If the temperature of the feed water entering the boiler is 100 deg. in equation (3): $x = 0.9758$ and $(1 - x) = 0.0242$; q for saturated steam at 362.8 deg. = 334.8; h for water at 100 deg. = 67.97; H for saturated steam at 362.8 deg. = 1194.4.

Then, the factor of correction is

$$F = 0.9758 + \frac{0.0242\,(334.8 - 67.97)}{1194.4}$$

or $F = 0.9812$.

Wear in Steam Turbines

It is frequently claimed for the steam turbine that its initial efficiency will be maintained whereas the steam engine, due to wear producing leakage of the piston and valve and derangement of the steam distribution, will rapidly deteriorate. S. Schulte, in *Glückauf*, says that the erosion of the blades of the turbine will produce changes in the direction and conditions of the entry of the jet which have a serious effect upon the efficiency. Temperatures over 250 deg. C. (482 deg. F.) may totally destroy bronze blades in from two to four years. Nickel-steel with 25 per cent. nickel has been found to be unusable. The best metal for blades is nickel-steel with 5 per cent. of nickel. The lateral wear of the bearings has also a detrimental effect upon the efficiency. The duration of the blade is considerably greater with lower steam velocity.

With a turbine of 500 kw. capacity, running at 3000 r.p.m., after 29,700 hours of operation with a production of 8,350,000 kw.-hr. the consumption had increased about 27 per cent.

A Curtis turbine of 1000 kw. capacity, running at 3000 r.p.m., with bronze blades, working at only one-half or three-quarters capacity, after 31,000 hours of running and a production of 17,800,000 kw.-hr. had an increased water rate of about 18 per cent.

In the case of a Zoelly turbine of 1000 kw. capacity, 19,500 hours of operation and a production of 13,160,000 kw.-hr. resulted in an increased consumption of about 3 per cent.

It would appear that in order to maintain the initial efficiency of a turbine, it may require occasional, if not frequent, reblading.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Catechism of Electricity

MANAGEMENT OF ELECTRICAL STATIONS

1304. *Is it not important to have system in the operation of an electrical station?*

Yes; because with a proper system each attendant has his special duties to perform, someone is responsible for whatever is to be done and the chance of overlooking matters is reduced to a minimum.

1305. *What provision should be made to safeguard attendants?*

Belts and moving parts of machinery should be inclosed with railings or guards, also dark pits and dark passages, such as back of the switchboard, must be well lighted, as shown in Fig. 428. "Stumbling blocks" should be removed or made as few as possible. Switchboards on which high-voltage currents are handled, should have an insulated space in the floor or an insulated platform in front of the board, as is also shown in Fig. 428. The same precaution should be taken with the generators, especially if the frames are not grounded.

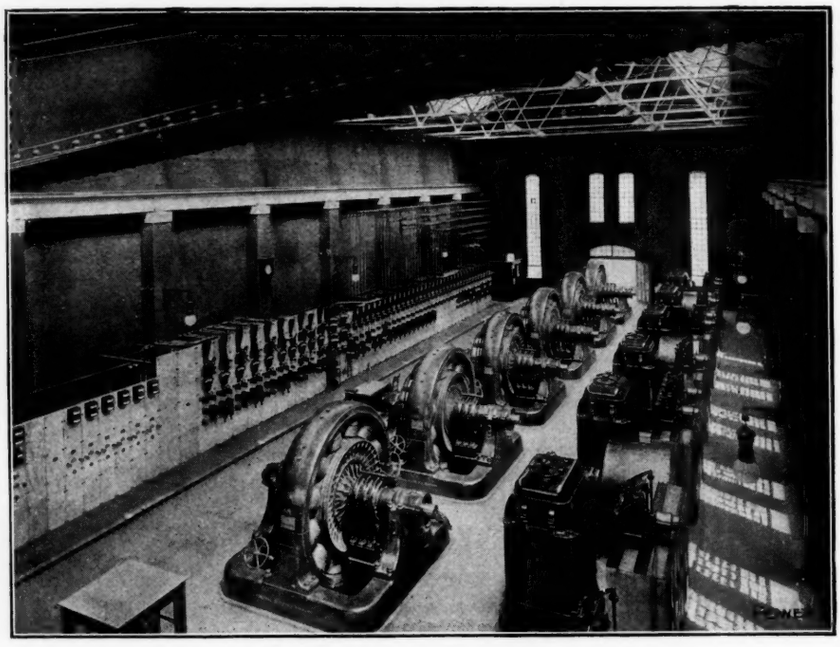


FIG. 428. SHOWING WELL LIGHTED PASSAGES BACK OF SWITCHBOARD AND TRANSFORMERS

1306. *Should the generator frames be grounded?*

Generators operating at a potential over 550 volts should have their base frames permanently grounded.

1307. *What advantage results from grounding a generator frame?*

Grounding a frame removes the possibility of dangerous shocks when coming in contact with it. The insulation of the entire system then depends upon the insulation of the generator conductors from the frame; if this breaks down, the system is grounded and the condition is indicated at once by the switchboard instruments.

1308. *Is there any argument against grounding a generator frame?*

Yes; the insulation resistance of the entire installation is reduced and there is an increased danger of shock from the line wires. This objection, however, can be overcome by employing a thorough system of insulation—a precaution which should obtain in any case, and which in consequence lessens the force of the argument against the practice.

1309. *How may a generator frame be properly grounded?*

By securely fastening one end of a wire to the frame and the other to a main water pipe inside the building. With a direct-connected generator an excellent ground may be had through the engine coupling and piping.

1310. *What precaution should be observed in connection with generators having grounded frames?*

High-potential generators with their frames grounded, should be surrounded by a small platform raised above the floor on porcelain or glass insulators so that those working about the machine may be protected from shock when adjusting the brushes.

1311. *How should generators be protected when not in use?*

Waterproof covers should be placed over them. Otherwise water may get into the working parts and cause an armature or field coil to burn out when the machine is started.

1312. *What provision should be made in the station wiring and apparatus to afford safety from fire?*

The station wiring and apparatus should conform to the rules and requirements of the National Board of Fire Underwriters. These are embodied in the "National Electric Code," a booklet which may be obtained free upon request from the electrical department of the National Board of Fire Underwriters, 135 William St., New York City.

1313. *What kind of fire-extinguishing apparatus should be placed in the station?*

Every station should be supplied with a fire pump and an adequate line of pipe and hose. The pump should be run at least once a week and kept in good working order. Care should be taken in locating the sprays and pipes so that water will not come in contact with the generators or the conductors. Also fire buckets filled with water should be placed in convenient places. Furthermore, it is important to have at hand a number of buckets filled with dry sand, which may be used in extinguishing fire between electrical conductors where water would do more harm than good. The buckets should be kept covered to prevent the sand from being blown around in the station.

1314. *What should be done to insure cleanliness in the station?*

The machines must be kept free from grease and dirt. Greasy rags and waste should not be thrown carelessly around but should be placed in cans provided for that purpose. Filing or repairing of any kind that produces dust or small particles of metal should, if possible, be performed elsewhere than in the generator room. Another aid to cleanliness and order is to keep all supplies under lock and key in a store-room.

1315. *How should the stock in the store-room be handled?*

If the station is large there should

be a person whose only duty consists in keeping charge of the supplies; and accurate accounts should be made by him of all receipts and disbursements. Applications for material should be made on regular printed forms which may be placed on file and used in keeping the stock books and accounts. In smaller stations, the store-room keeper may perform other duties as well.

1316. *What is the best method of issuing orders to the station attendants?*

Having them plainly written and fastened by means of thumb tacks upon a bulletin board in a conspicuous part of the station. All operating orders, such as the dynamo schedule, circuit schedule and orders for changes, should be handled in this manner. Each dynamo and each circuit should have its number, and the orders should make it unmistakably plain as to when each dynamo shall start and stop; also during what time and from which machine, each circuit shall be operated. There should always be room left on the board for a notice to be posted when a high-voltage line is being worked on, and when a man is in a flywheel pit or in any other position where he is liable to be injured by the starting of some particular machine.

1317. *How are reports of value in the operation of a plant?*

Without accurate reports an engineer cannot keep a check on what is being done.

1318. *What is the best way of collecting material for the reports?*

By means of recording instruments and keeping all the paper dials or cards obtained from them on file. Most of the necessary calculations may be made from these records. When the station is not provided with recording instruments, the different meters in service should be read at least every half hour and the readings entered on regular report blanks. In large stations there should be separate blanks for the different departments, but in a small station, one blank may be used for all.

1319. *Give an example of a common form of power-house report.*

Fig. 429, represents a power-house report on four machines for one day, from midnight to midnight, the generators being driven by two waterwheels. Two of the generators are 550-volt direct-current machines, the third is a 2000-volt alternator having a capacity of 75 amperes, and the fourth an arc-light generator carrying a load of forty-five 2000- cp. arc lamps. There are but three circuits, the two direct-current generators being run in multiple when the load exceeds 100 amperes.

1320. *What information regarding the operation during the 24 hr. may be obtained from this report?*

In the vertical columns are recorded the readings on the instruments of each machine, at the end of every half hour. At the end of a day's run, the columns are added and averaged; the values are then recorded in the upper right-hand corner, as shown. The voltmeter for the alternator is connected across the secondary wires of a transformer that reduces the pressure to 100 volts when the current is generated at 2000 volts; in other words, it transforms the voltage in a ratio of 20 to 1. In making out the report, the voltage across the secondary terminals of the transformer is recorded. The arc-light voltage across the terminals at the switchboard is recorded and indicates, approximately, the number of lamps burning when the readings are being taken. This voltage is usually entered in round figures, that is, to the nearest multiple of 50—the voltage per lamp. The average voltage is treated in the same manner.

1321. *How are the calculations relating to the individual generators arrived at?*

At the end of the day the average current from each generator is multiplied by its average voltage, giving the average watts developed by the machine. This multiplied by the number of hours it has been in operation and divided by 1000 gives the output of the dynamo for the day, in kilowatt-hours. By adding together the outputs of the machines, the total work done by the plant will be found, which, in this case, is 1846 kw.-hr. This may be reduced to horsepower by multiplying by 1.34.

THE POWERVILLE ELECTRIC CO.

POWER HOUSE REPORT

TUESDAY, JAN. 16, 1911

TIME.	DYNAMO NOS. 1 A	1 V	2 A	2 V	3 A	3 V	4 A	4 V
12.30	40	500			20	103	10.2	1250
1.00	30	505			18	103	10.2	1100
1.30	30	500			15	102	10.5	1000
2.00	20	500			12	102	10.5	1000
2.30	40	500			10	101	10.5	1000
3.00	20	500			14	102	10.4	1050
3.30	15	500			28	104	10.3	1150
4.00	50	510			32	104	10.2	1500
4.30	60	525			37	105	10.2	1850
5.00	65	540			37	105	10.2	1850
5.30	80	540			25	103	10.1	1950
6.00	110	525			23	103	10.5	1000
6.30	80	530	45	530	20	103	10.5	1000
7.00	70	530	70	530	16	102		
7.30	100	550	100	550	14	102		
8.00	100	550	90	550	9	101		
8.30	90	550	90	550	10	101		
9.00	95	550	85	550	10	101		
9.30	80	550	75	550	15	102		
10.00	60	540	65	540	14	102		
10.30	65	540	65	540	16	102		
11.00	80	545	65	545	16	102		
11.30	80	545	70	545	15	102		
12.00	90	550	85	550				
12.30	90	550	85	550				
1.00	90	550	80	550	12	102		
1.30	85	550	85	550	14	102		
2.00	80	550	75	550	14	102		
2.30	75	550	75	550	14	102		
3.00	75	540	60	540	15	102		
3.30	70	540	60	540	16	102		
4.00	80	550	75	550	20	103		
4.30	80	555	80	555	28	104		
5.00	100	555	90	555	39	105	10.5	1050
5.30	105	560	105	560	58	108	10.3	1200
6.00	100	560	110	560	70	109	13.1	1950
6.30	90	550	110	550	75	110	9.8	2250
7.00	90	550	95	550	78	110	9.8	2250
7.30	80	550	90	550	76	110	9.8	2250
8.00	80	540	85	540	75	110	9.8	2250
8.30	60	530	80	530	73	110	10.2	2200
9.00	50	540	70	540	73	110	10.2	2200
9.30	40	530	65	530	65	109	10.2	1850
10.00	40	530	50	530	55	107	10.2	1700
10.30			85	555	40	105	10.2	1400
11.00			45	535	38	105	10.2	1400
11.30			45	525	29	104	10.2	1350
12.00			40	500	25	103	10.2	1250

Dynamo Nos.	Started	Stopped.	Hrs. Run.	Averg. Amps.	Averg. Volts.	Kilowt. Hours.
1	12.00 p.m.	10.00 p.m.	22.0	70.7	536.4	834
2	6.30 a.m.	12.00 p.m.	17.5	76.3	543.8	726
3	12.00 p.m.	11.30 a.m.				
	1.00 p.m.	12.00 p.m.	22.5	31.0	104.2	73
4	12.00 p.m.	6.30 a.m.				
	5.00 p.m.	12.00 p.m.	13.5	10.2	1550	213
Wheel No. 1	12.00 p.m.	11.00 p.m.	23.0			
" 2	4.00 a.m.	12.00 p.m.	20.0			

Average No. Arc Lamps..... 31
Average No. Incandescent Lamps..... 1,240
Total No. Incandescent Lamp-hours..... 27,900
" " Kilowatt-hours..... 1,846
Head of Water, 6 a.m., 27 ft. 4 in.; 6 p.m., 27 ft. 6 in.

SUPPLIES RECEIVED.	SUPPLIES USED.
31 gal. cylinder oil.	1 gal. cylinder oil.
31 " Renown engine oil.	2 " Renown engine oil.
4 doz. dynamo towels.	1 " O.K. dynamo oil.
6 " 10-in. files.	2 cans potash.
	1 10-in. file.

REMARKS.

Short circuit on dynamo No. 1, from 6.10 a.m. to 6.20 a.m.
Continued heavy ground on dynamo No. 3 Occasional poor contact in arc circuit.
Dynamo No. 2 running exceptionally warm

Signed.

JOHN BROWN.

FIG. 429. FORM OF POWER HOUSE REPORT

1322. *Why is it necessary to record the average number of arc and incandescent lamps that are burning during the run?*

Because the revenue of a lighting company is normally figured on this basis.

1323. *How is the average number of arc lamps calculated for the report?*

The number of arc lamps is directly proportional to the voltage of the arc dynamos; therefore, dividing the average

pressure, 1550, by 50, which may be taken as the voltage of each lamp, the result will be approximately the average number of lamps burning during the run of 13½ hr., which in this case is 31.

1324. *How is the average number of incandescent lamps calculated for the report?*

These may be calculated from the current of the alternator. The 16-cp. lamp is taken as the standard, and it is customary to allow 50 watts for such a lamp. If the voltage is 100, each lamp will consume ½ ampere, and, since the ratio of the transformer is 20: 1, 20 amperes in the secondary winding will result from a current of 1 ampere in the primary; consequently there will be 40 lamps (20 ÷ ½) burning in the secondary circuit for each ampere of current generated by the alternator. The machine was delivering

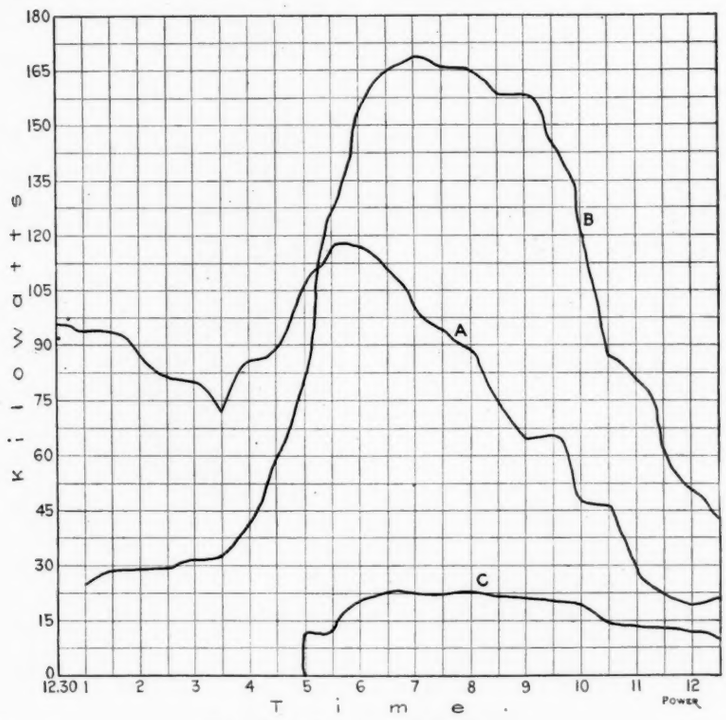


FIG. 430. TYPICAL LOAD CURVE

on an average 31 amperes, making the average number of incandescent lamps

$$31 \times 40 = 1240$$

Multiplying this number by 22½ hours during which the alternator was running, shows the number of lamp-hours to be 27,900.

1325. *Is it not sometimes more desirable to know the total kilowatt-hours supplied to the different parts of a system, than to know the average number of lamps burning?*

It may be; in which case this part of the report should be changed to suit conditions.

1326. *What information belongs under the heading "Remarks"?*

Notes regarding the general condition of the plant and of the individual machines, especially when anything out of the regular order occurs. It is especially important to give an accurate and minute description of all accidents, together with their causes. Carefulness in this matter often enables the engineer to shake off responsibilities that are liable to be heaped upon him by other parties.

1327. *What conditions besides those previously mentioned will affect the general makeup of the report?*

When there are many different circuits, the time when they are opened and when they are closed should be noted, together with the conditions of the circuits. When this is done, it is best also to note the state of the weather, since that greatly affects the condition of the lines.

The report shown is for a water-power plant. When steam is used, a report of the boiler room, etc., is usually included. An extra column is also provided for a record of the steam pressure, and space must be left to show when the fires are started, when the boilers are cleaned, the amount of fuel received and consumed, and further data necessary to make a complete report. Here, again, a variation may take place. A separate report may be kept of the steam plant, which may be done by an engineer who has nothing whatever to do with the dynamos, etc. This method is generally adopted in large plants, but in small plants it is not necessary and will only cause extra labor.

1328. *How often should power-house reports be turned in to the office?*

In most companies it is required that a report be sent to the office each day; but sometimes only the general information regarding load and accidents is wanted, in which case it is advisable to always make out a full report as explained, and then make a synopsis of this for the office.

1329. *What is the best method of furnishing the office with information regarding the loads carried by the different machines?*

Plot the readings on a sheet of coordinate paper, as shown in Fig. 430.

1330. *Explain how the graphical report in Fig. 430 furnishes information regarding the loads on the generators.*

Curve *A* represents the output of dynamos Nos. 1 and 2, curve *B* the output of the alternator and curve *C* the output of the arc-lamp dynamo. The height of the curve *B*, for instance, at 8 p.m. shows the output of the alternator at that moment to be 165 kw. When desired a curve may be similarly drawn showing the combined output of all the generators.

1331. *Does the management of a substation differ from that of a central station?*

Only in minor details. Rotary converters and synchronous motors, as has previously been explained, are started somewhat differently from ordinary generators and they are not handled in exactly the same way when they are brought to rest. They are, however, taken care of and repaired in the same manner as generators. The principal thing to observe is that these machines must not be thrown onto the line unless they are in step with the apparatus supplying the current. They must also be protected with reliable safety devices that will open the circuit if they should be pulled out of step.

Electrically Heated Office Building

Cheap hydro-electric power, generated on the Snake River, Idaho, has helped develop the city of Twin Falls very rapidly and has even rendered practicable the electric heating of buildings, says the *Scientific American*. In place of steam radiators, electric heating units have been installed in the vicinity of Twin Falls to the extent of nearly 1000 kw. connected load. A six-room house having eight outlets uses $70 to $100 worth of electrical energy during the season of eight months. A building having stores and offices on the first floor and a ballroom on the second floor uses $625 worth of electrical energy for the season.

Three-phase current is brought into this building through an underground conduit at 2300 volts and transformed to 220 volts for the 40 electric radiators and a water-tank heater, which keeps water continually at the boiling point through thermostatic control. The size of the building is 50x120 ft., and the electrical equipment cost about $1000, exclusive of the cost of the transformers, which were furnished by the power company.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Oil Engines

The current report of the "prime movers" committee of the National Electric Light Association contains interesting data upon the fuel-oil production and the status of the oil engine in this country. The following statistics for 1909, covering the petroleum production and the amount of oil available for power purposes, are given:

	Barrels	Per Cent.
Total production (crude).....	182,000,000	100
Exported (crude).....	5,184,700	2.8
Domestic consumption.....	176,815,300	97.2
Total amount available for gas and fuel oil.....	68,264,000	100
Exported (gas and fuel oil).....	2,901,000	4.2
Domestic consumption (gas and fuel oil).....	65,363,000	95.8
Oil burned in locomotives.....	20,000,000	29.3
Used in gas making.....	13,184,000	19.3
Available for power.....	32,179,000	47.2

The report states further that, although figures on the oil distribution for 1911 are not yet available, it is estimated that the total production in the United States last year was 209,500,000 bbl., of which 77,800,000 bbl. were produced west of and 132,200,000 east of the Rocky Mountains. While the total production showed an increase over 1910, the Eastern oil fields showed a slight decrease. This was offset, however, by an over-production in excess of the consumption in California of 10,500,000 bbl. This had a tendency to stop further drilling.

Fuel oil as supplied for power purposes consists ordinarily of a distillate that comes from the distillation of crude petroleum after the gasoline, kerosene and lubricating oils have been abstracted. It distills at temperatures from 300 to 700 deg. F. and varies in specific gravity from 40 to 20 deg. Baumé. The modern oil engines have been able to handle fuel oils successfully that are as heavy as 17 deg. Baumé and as light as kerosene, or 40 to 48 deg.

The heat value of oil varies from about 18,000 to 20,000 B.t.u. per pound and this seems to bear an almost direct relation to the density. In Germany as well as in the United States gas-house tar has been used as fuel in engines of the Diesel type, but its use has not proved entirely satisfactory as it is impossible to completely free it from water, which is apt to cause the engine to misfire and give poor speed regulation.

The character of the fuel oil is not always dependent upon the nature and quality of the original crude oil, as the methods of distillation are such that a uniform grade of residual oil is not always produced. However, the various grades of fuel oil, as now supplied for power purposes, give satisfactory results if free from objectional ingredients, such as sulphur, moisture and acids, the acid being an ingredient mixed with the oil during its purification after distillation. Sulphur, in the form of SO_3 or CS_2, ordinarily in a dry gas produces no harmful results, but in the presence of moisture sulphuric acid is formed and this attacks the pistons, cylinders and valves.

In the language of the committee, the oil engine differs from the more familiar gas engine only in that the fuel must be prepared for combustion at the engine, or injected at high pressures, and therefore either a carburetor, vaporizer or an injection valve must be provided for vaporizing the liquid-oil fuel, depending upon whether low, high or modified compression types of engines are used. The simple automobile engine comes within the former class (low pressure). For power purposes the low-pressure engine frequently employs a hot-bulb or "preheating chamber," to enable the engine to burn comparatively low-grade oils. The Diesel engine employs high-combustion pressures (500 lb., approximately) and requires an air-injection pressure of about 1000 lb. It has a cycle peculiar to itself, and the distinguishing feature is that the fuel is injected at constant pressure for about 10 per cent. of the working stroke, while the air is hot enough by reason of the high compression to cause combustion on contact.

With the fuel admitted only after the working stroke begins, there is no opportunity of a premature explosion to take place. Moreover, as only a small clearance of about 7 per cent. is necessary, more complete scavenging may be effected, which improves the efficiency accordingly.

The long-sustained high pressure and temperature of the regular Diesel engine obviously places some severity upon the operating parts, thereby requiring sturdy construction. On the other hand, the suddenly applied pressure in the low-pressure type, Beau de Rochas or Otto cycle, introduces high stresses which must also be provided for.

In point of fuel economy the superiority of the Diesel type has been very well established. Regarding maintenance and attendance, the low-pressure type has much in its favor. This would lead to the opinion that for the larger sizes, exceeding 150 hp., the Diesel motor should be chosen, while in case of small power requirements (under 100 hp.) the low-pressure type would generally be used. In the neighborhood of 100 to 150 hp. the field will be disputed between the two according to existing fuel prices and local conditions. There has been developed a design which may be termed a "compromise" between the low-pressure hot-bulb and the Diesel high-compression engines, for which there is claimed a fuel economy comparative with the Diesel motors, although it employs lower working pressures.

The guarantees on oil engines ordinarily offered by the manufacturer in the past have been as follows, for loads between three-fourths and full-load rating:

Diesel

Oil consumption not to exceed 0.5 lb. per horsepower-hour.

De La Vergne

Oil consumption not to exceed 0.6 lb. per horsepower-hour.

Hornsby-Akroyd

Oil consumption not to exceed 1 lb. per horsepower-hour.

Oil not to contain less than 18,500 B.t.u. per pound.

In response to inquiries sent by the committee to a large number of plants many interesting replies were received relative to the operation, maintenance and cost in American plants using oil engines. These are herein tabulated.

The load factor in the table is taken as the average percentage of rated load carried by the engine during its period of service.

The operators registered several criticisms of the oil engines regarding minor troubles, but in no case was there any serious complaint made, with the exception of a few instances where breakages were directly traceable to the inexperience of the operators, or to the possible inadequacy of the earlier designs. Pistons and piston rings have been most prominent in the difficulties experienced with the engines, more particularly with the high-pressure type. The carbonization and consequent sticking of the piston rings on either very light or very heavy loads is a constant source of trouble, as is also the rapid wearing down of rings and cylinders, necessitating reboring about every three years, and reported as due either to faulty lubrication or lack of a crosshead in the use of trunk-type pistons.

Character of Load	Date, Units Put in Operat'n	Number of units	Individual b.hp.	Type	Vertical or horizontal	No. of cylinders	Diameter of cylinder and length of stroke	Rated speed, r.p.m.	A.c. or D.c.	Load factor, per cent.	Kind of Fuel and Source	Sp.gr., deg. Bé.	B.t.u. per lb.	Price of fuel, cents per gal.	Operating Cost { Cents per Kw.-hr. / Cents per B.hp.-hr. } Fuel	Labor	Lubricants	Supplies	Repairs and Maintenance	Total	Character of attendance necessary	Trouble	Remarks
Central station	1905	3	225	Diesel	V	3	16x24	164	D.c.	50	Penn. fuel	30	19,832	3.0	0.33	0.3	0.05	0.01	0.3	0.99	Best	Some	
Central station	2–1907 2–1910	4	225	Diesel	V	3	16x24	164	A.c.	95	Fuel	19	18,000	3.5	0.42		0.024				High grade	Yes	
Industrial, 55-hr. week†	1910	1	170	De La Vergne	H	2	17x27½	180	A.c.	65	Penn. fuel	29	19,000	2.9	0.198	0.28	0.056				Medium	None	
Industrial, 10-hr. day†	1911	1	125	De La Vergne	H	1	20x34½	155	D.c.	70	Texas fuel	27	19,500	3.0	0.205	0.4					Medium	None	Variable load
Industrial, 8-hr. day	1911	1	225	Diesel	V	3	16x24	164	D.c.	70	Texas fuel	27	19,900	3.0	0.4	0.306	0.044			0.75	Medium	None	
Refrigerating and lighting	1911	2	120	Diesel	V	3	12x18	225	A.c.	60	Kansas fuel	34	19,600	2.75	0.28	0.16	0.09		0.003	0.533	Best	None	
Central station*	1909	3	450	Diesel	V	3		150	A.c.	75	Mexican gas oil	38		3.8	0.364	0.321	0.076	0.055	0.152	0.968	High grade	None	
Industrial, 10-hr. day	1909	3 1	500 170	De La Vergne	V H	3 2	17x27½	180	D.c.	54	Penn. residual	26	19,500	2.25	0.236	0.371	0.061	0.034	0.151	0.853	Medium	None	
Industrial, 10-hr. day	1910	1	85	De La Vergne	H	1	17x27½	165			Crude										Medium		
Industrial, 11-hr. day	1911	1	125	De La Vergne	H	1	20x34½	155	A.c. D.c.		Fuel			3.15							Medium	Yes	
Oil pumping		1	125	De La Vergne	H	1	20x34½	155															
Industrial, 11-hr. day	1905	2	75	Diesel	V	3	10½x15	250	D.c.	40	Illinois crude	30	19,000	2.3	0.27	0.413	0.071	0.019		0.773	High grade	None	
Central station	2–1905 1–1907	2 1	225 120	Diesel	V	3	16x24 12x18	167 225	A.c.	41	Illinois fuel	40	19,500	2.73	0.26	0.207	0.04	0.055		0.562	Medium	Some	
Central station, 20-hr. day		2	225	Diesel	V	3	16x24	164	A.c.	80	Fuel			3.0	0.38	0.29	0.03	0.11		0.81		Yes	
Industrial, 54-hr. week	2–1906 2–1910	4	225	Diesel	V	3	16x24	164	A.c.	73	Penn. fuel	29	19,000	3.1	0.295	0.144	0.045	0.01	0.217	0.711	Medium	Some	
Refrigerating		2	225	Diesel	V	3	16x24	164															
		1	120		V	3	12x18	220	D.c.		Fuel	28	19,000	4.0									
Irrigation, pumping		1	180	De La	H	2	17x27½	200			California crude	17	18,500								Medium	None	
		1	260	Vergne	H	2	20x34½	164															
Industrial, 10-hr. day	1–1908 3–1909 2–1910	6	225	Diesel	V	3	16x24	164	A.c.	83	Penn. fuel	28	19,443	2.75	0.283	0.176	0.084	0.07	0.09	0.703	Medium	None	
Central station†	1911	1	170	Diesel	V	3	14x21	200	A.c.	25	Missouri fuel	19	18,000	3.56	0 339		0.049				Medium	None	
Central station	1910	1	225	Diesel	V	3	16x24	164	A.c.	28	Illinois crude	25		1.77	0.219	0.484	0.075	0.002	0.05	0.83	Medium	Some	
Mining plant†	1911	1	125	De La Vergne	H	1	20x34½	155		64	Kansas fuel	27	19,000	13.0	0.778	0.468	0.078	0.087		1.411	Medium	Some	
Industrial, continuous	1907	3	225	Diesel	V	3	16x24	168	A.c.	80	Fuel	31		2.5	0.245	0.161	0.03		0.07	0.506	Best	Yes	Engines overhauled every 10 days
Industrial		1	15	Hornsby	H	1	12x16	260						2.5							High grade	None	
Pumping		1	25	Ackroyd		1	14x20	220						3.5							High grade	Yes	Steam plant in reserve
Industrial, continuous†		16	125	H.-A.	H	1	27x36	150	D.c.	75	Texas fuel	27	19,000	2.66	0.293	0.15	0.05		0.2	0.693	High grade	Yes	
		6	225	Diesel	V	3	16x24	165							0.184	0.15	0.06		0.25	0.644	High grade		
Central station, 131-hr. wk	1911	1		De La Vergne										2.0							Medium	None	
Industrial, continuous	1908	16	225	Diesel	V	3	16x24	164	A.c.	73				2.3	0.189	0.121	0.032	0.01		0.352	High grade	Very little	
Pumping, continuous	1910	1	125	De La Vergne	H	1	20x34½	155		32	Colorado fuel	32	19,530	3.0							Medium	None	
Industrial, 10-hr. day		1	85	De La Vergne	H	1	17x27½	164	A.c.	85	Penn. fuel			3.25							Medium	Very little	
Industrial		6	225	Diesel	V	3	16x24	164	A.c. D.c.		Texas fuel	29	19,500								High grade	Very little	
Central station	1907	2	225	Diesel	V	3	16x24	164															
Central station	1910	1	120	Diesel	V	3	12x18	225	A.c.		Indiana fuel			3.0							Medium	None	Auxiliary to steam plant.
Central station	1906	1	170	Diesel	V	3	14x21	200	A.c.		Penn. fuel	29	19,000	3.5							High grade	None	
		1	225				16x24	164															
Industrial, 10½-hr. day		1	225	Diesel	V	3	16x24	165	D.c.	48	Illinois fuel	30	19,200	2.2	0.25	0.55	0.05	0.32	0.14	1.31	Medium	None	Variable load
		1	120				12x18	220															
Industrial	1910	1	90	De La Vergne	H	1	17x27½	180		78	Illinois fuel	29	19,000	3.5							Medium	Some	
Oil pumping	1911	1	85	De La Vergne	H	1	17x27½	180			Illinois fuel	32	19,100										
Central station	1905	2	120	Diesel	V	3	12x18	225	A.c.	50	Illinois crude			2.0	0.25	0.33	0.083		0.5	1.163	High grade	Very little	
	1910	1	225				16x24	170															
Industrial, continuous	1910	2	195	Diesel	V	3	14x21	200	D.c.	60	Illinois fuel	29	19,000	4.0	0.501	0.278	0.103		0.049	0.931	High grade	Some	

*The 450- hp. engines were built by the Ausgburg Co. and the 500-hp. engines were built by Sulzer Bros. †The costs have been given on the b.hp.-hr. basis; all others are per kw.-hr.

The low-pressure type has been more or less free from difficulties but has suffered from carbonization of oil in the vaporizer, as well as a tendency of the engines to loosen on their foundations. In either type of oil engine, where the engine has more than one cylinder, great care must be exercised in correctly aligning the crankshaft and equalizing the work done by each cylinder. If these points are not regularly checked, liability of springing or damaging the crankshaft exists. In practically all cases the regulation has been reported to be satisfactory. Where the low-pressure oil engine is installed there does not seem to be the same stress placed upon the necessity for high-grade attendants. This is possibly due to the character of service deemed satisfactory where they happened to be used. In plants with engines of the high-pressure type it appears to be an almost unanimous opinion that skilled operators should be employed. The valves must be frequently reground to prevent loss of efficiency and capacity through leakage. There are other small adjustments which require attention to maintain the original quality of the engine.

National Gas Engine Association Convention

The convention of the National Gas Engine Association, held in Milwaukee, formally opened on Wednesday, June 19, with an address of welcome by Mayor Bading, of that city, and a response in behalf of the society by H. W. Bolens, of Port Washington, Wis.

Prominent in the morning's session was an address by P. S. Rose, of the publicity committee, who called attention to the possibilities of the conservation of our food supply through the displacement of farm-work animals by gas engines. He showed that fully one-fourth of the tillable land in the United States is given over to the support of horses and mules, useful for power alone, and emphasized the fact that as the gas engine displaces the horse, this land becomes available for agricultural purposes.

Two formal papers were read during the morning session, "The Adaptability of the Large Gas Engine in the City," by H. W. Jones, vice-president of the association, and "Some Lubrication Problems," by A. E. Potter, of New York. Mr. Jones gained instant attention by the statement that there were more gas engines built in Wisconsin last year than there were steam engines built in the whole United States during the same period. His paper was largely a plea for greater publicity regarding the economy and reliability of the gas engine and that its use completely solves the smoke nuisance.

Mr. Potter's paper was quite technical; he analyzed the difficulties of lubricating the various parts of different types of engines and dwelt on the characteristics of various oils. Interesting among his conclusions was that a gas-engine cylinder oil may be of too high a fire test, failing to burn in the combustion chamber, accumulating and fouling the ignition system and causing misfiring. He recommended the sale of oil to the consumer in sealed packages bearing trade names.

In the discussion which followed, it was stated that most engine manufacturers recommend "any good gas-engine oil" and leave the customer to struggle with the lubrication problem, a practice which Mr. Potter unqualifiedly condemned.

Following the presentation of these papers, E. L. Tracy, formerly state oil inspector of Wisconsin, spoke on the methods used in the state inspection. He contended that the determination and certification of the specific gravity of gasoline or oil protected nobody; that large purchasers could obtain the information much cheaper themselves and small consumers had to buy what was offered them, anyway. He showed that the inspection in all the Central and Western states was similar to that in Wisconsin, furnishing only fees for political purposes. After Mr. Tracy's talk, resolutions were adopted by the association condemning the inspection system now in vogue as a useless burden on the industry which the consumer had to carry.

Wednesday evening the regular banquet of the association was held in the fern room of the Hotel Pfister. A pleasant feature of the evening was the presentation of a pair of field glasses to O. M. Knoblock, treasurer of the association, the occasion being his birthday.

On Thursday, four formal papers were presented: "The Storage Battery in Isolated Lighting Plants," by Taliaferro Milton; "Oil Power," by Edward T. Adams; "Dirty Spark Plugs and Missed Explosions in the Sales Department," by Frank J. Whiteside, and "Electric Farm Lighting Systems," by Otto Borcherdt.

Mr. Milton called attention to the many uses for power about large country estates and advocated the installation of central electric plants with electric drive for the individual machines. A storage battery gives 24-hr. service for lighting and will also serve to run small motors when the main plant is shut down, resulting in increased convenience and economy and decreased fire risk as compared with direct gas-engine driven machines.

Mr. Adams analyzed the oil and gasoline situation, which is fast becoming acute, and showed that kerosene must soon supply the demand for power which gasoline has created and that we are even now entering the age of oil power.

A valuable point brought out by Mr. Whiteside in his paper on defects in selling organizations, was that too frequently letters of inquiry are not carefully read and answered, resulting in disgruntled customers and lost business. The meeting of Friday opened with a general discussion on how to best help the dealer increase his sales, during which the dominant and fraudulent advertisement were both severely scored. The day's program was given up to publicity and sales increase, the papers presented being: "Hit and Miss Publicity," by E. J. Baker; "Publicity through the Trade Journals and What It Means to the Manufacturer," by E. R. Shaw; "Essentials of Foreign Trade Development," by Franklin Johnson; "Direct Advertising to the Dealer," by C. H. Hall, and "The Small Farm Tractor," by L. W. Ellis. Mr. H. E. Miles also spoke on industrial education. The alarming growth of illiteracy in our native white population and the present state of social unrest moved Mr. Miles to advocate continuation schools, somewhat after the German type, in connection with factory organizations. This he pointed out would make for skilled and efficient workmen as well as better citizens. Before adjournment the association adopted resolutions asking the Post Office Department to modify its attempted ruling requiring technical magazines to be shipped by freight.

The Gas Engine Show

In connection with the convention there was held in the auditorium the first annual gas-engine show of the association, about seventy-five manufacturers of gas engines and accessories taking advantage of the occasion to display their products. The show was unique in being the first of the kind where engines could be run under normal operating conditions, making possible for the first time the satisfactory demonstration of systems of starting and control. Engines aggregating about one hundred cylinders were in operation and it is a tribute to the present perfection of the gas engine that so many engines could exhaust into a single room and the room remain habitable as a place of exhibition.

A tugboat engineer visited the engine room of a large ocean steamer. One of the engineers on watch showed him around the engine room and explained the operation of various devices. Stopping at the gage board, where all of the needles were pointing to zero, the visitor looked at the vacuum gage and said:

"What vacuum do you carry?"

"Anywhere from 24 to 26 in.," the engineer replied.

"Gee!" said the tug man, "is that all?"

"How much did you think we ought to carry?"

"Why we carry 24 on my little tug, and I thought a big ship like this would carry at least 40."

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Friction Formulas for Commercial Pipe*

By Ira N. Evans

To determine the best formula for the friction of water in commercial wrought-iron pipe as used in heating work, the results from several standard formulas are compared in the accompanying curves and tables. The results were figured analytically and plotted for sizes from 1 to 12 in. and velocities from 1 to 15 ft. per second inclusive.

Most of these formulas were based and developed on water-works practice for cast-iron pipe where a considerable increase in head is contemplated due to the tuberculating action of the water on the interior surface. Comparatively few tests have been made on wrought-iron pipe below 8 in. in diameter. Tests on galvanized-iron pipe show an especially high friction factor even when the pipes are new. Wrought-iron pipe has a comparatively low friction factor, and the results of the formulas vary between these limits. The greatest variation is found in small pipes, such as are most used in heating work. Here water has no action on the pipes if the piping is tight and the water changed infrequently. The high velocities and continuous service throughout the heating season tend to improve the surfaces and reduce the loss in friction head.

Table 1 gives the results from the several formulas so arranged that the loss in head for each size and velocity may be read according to each authority. The actual commercial diameters are used. Table 2 gives the discharge in cubic feet, gallons and pounds per minute for each size and velocity. The temperature of the water is taken at 100 deg. F. and the pounds and gallons per cubic foot as 62 and 7.48 respectively. Table 2 is probably not closer than the third or fourth figure to the actual results, due to the decimal expressing the area in square feet.

The curves are plotted from Table 1 and the averages of all results are given in the table and also plotted. In addition to the results obtained from the formulas several actual tests have been plotted and averaged for different sizes. At a glance the curves show which formulas are the closest to the average for the various sizes and velocities.

For hot-water heating by forced circulation the average of these formulas should give a safe and proper friction head. Each formula represents the best results of a long series of investigations by that particular authority so the average of all should be very nearly correct. Some of these formulas give high results for small sizes and low for large, and *vice versa*. The curves are differently related as the pipe sizes increase, the widest variation occurring with the high velocities.

The two tests by Poché on 1-in. wrought-iron and galvanized-iron pipes were evidently carefully made and no doubt are very accurate. Each test coincides exactly with one of the formulas.

A study of these curves in connection with some of the published matter on this subject will show the danger of using a table of friction heads and discharge quantities in designing work without first investigating and checking the data. Curves plotted to small scale cannot be read accurately so that anyone having continuous use for these data should work out tables independently. It will then be possible to judge the weight of each factor and use the data with confidence of obtaining the proper results for the governing conditions.

Tabular data and charts giving the discharge and friction head for certain pipe sizes are likely to lead to error, due to the many factors involved, unless these factors are specifically expressed. The plotting should be between friction head and velocity in feet per second. The actual diameter instead of the nominal does not affect the friction head as it does the discharge. If this suggestion is not followed it is almost impossible to check the curves and tables except by trial and error.

The author had two or three seemingly good charts on this subject, but they were useless as the plotting was between discharge in gallons and friction head. An article by J. N. LeConte in the *Journal of Electricity, Power and Gas* for Jan. 27, 1912, purporting to give a series of friction factors from Williams & Hazen's formula to be used in the old formula

$$h = f\frac{l}{d}\frac{V^2}{2g}$$

gave the author a great deal of trouble. The Williams & Hazen coefficients gave very high results when plotted, and when the original data were looked up it was found LeConte had used a friction factor of 100, which was for cast-iron pipe 17 years old and utterly unsuited for the purpose of this article.

Several have claimed that the viscosity of water affects the friction head and it may, but the roughness of the surface of the pipe seems to vary results more than any other factor and is the one most difficult to determine for any given case. It requires assuming a constant to fit the probable roughness of an unknown pipe.

Some of the tests given were conducted by Saph and Schoder, at Cornell University, but were on old pipes and galvanized in some cases, which made the results high. Those by Poché were published in Power, July 25, 1911, and were on pipes in perfect condition. The results are therefore nearer the requirements for the conditions under consideration.

Old tuberculated and galvanized pipe show such variable and uncertain results that the data obtained are practically useless for the purpose in hand. Anyhow, in hot-water heating work, if the piping deteriorates to any extent, it does so rapidly, manifests itself in leaks and has to be replaced.

Some authors have taken the liberty of varying the friction factor according to their personal judgment, and the reader using it sometimes errs again in adding an additional safety factor to be on the safe side. The friction factor may also vary greatly if care is not used in the erection of piping. The ends of pipes entering fittings and couplings should be carefully reamed to remove any burr left in cutting and threading.

In hot-water heating, as far as design of mains is concerned, the friction factor is not so important if the formula gives results which are not high for one size and low for another for changes in velocity. Using a high factor would not especially affect the circulation of the different circuits as they would still balance. The designs of the pumps and prime mover are, however, affected. If the head of the pump is greater than required for the system for a given discharge, the latter will be increased somewhat and the prime mover, if a motor, may be overloaded.

Formula 1, that of Williams & Hazen, seems to coincide with the average values throughout nearer than any of the others.

$$V = cr^{0.63}\left(\frac{h}{l}\right)^{0.54} 0.001^{-0.04} \qquad (1)$$

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TABLE 1. FRICTION HEAD PER 100-FOOT OF PIPE

Size Pipe, In.	Formula or Test	Velocity Ft. per Second														
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
	Poche, galv. iron	1.17	2.83	6.16	10.83	16.7	23.3	31.3	40.2	50.05	59.9	70.2	81.0			
	Poche, wrought iron	0.5	1.82	4.16	7.17	10.14	13.09	19.41	25.0	31.00	37.53	45.6	53.3			
	Fanning	0.675	2.3	4.68	7.95	12.17	17.13	22.8	29.3	36.5	44.6	53.3	62.9			
	Meier	0.85	3.1	6.5	12.00	16.7	23.54	31.36	40.2	50.05	60.89	72.7	85.5			
	Harrison	0.45	1.81	4.08	6.83	10.2	14.3	19.0	24.5	30.5	37.3	44.7	52.8			
1	Sercinsky	0.8	2.6	5.75	10.00	14.95	20.8	27.5	36.00	45.00	55.0	66.00	75.0			
	Darcy	0.691	2.8	6.22	11.05	17.3	24.9	33.85	44.2	55.6	69.0	83.6	99.5			
	Williams & Hazen	0.75	2.7	5.7	9.72	14.69	20.6	27.4	35.14	43.63	53.0	62.35	71.86			
	Sapp. Schroder, old galv.	0.76	3.04	6.84	11.9	18.4	26.1	34.8	45.5	56.7	70.0	82.5	97.0			
	Average	0.74	2.6	5.59	9.72	14.6	20.42	27.5	35.56	48.86	54.15	64.55	75.4			
	Fanning	0.5	1.8	3.54	6.01	9.19	12.94	17.28	22.14	27.7	33.6	40.4	47.48			
	Meier	0.6	2.17	4.65	7.8	11.84	16.7	22.3	28.5	35.6	43.3	51.6	60.7			
	Harrison	0.3	1.09	2.44	4.34	6.79	9.77	13.3	17.4	21.98	27.14	32.84	39.1			
1¼	Sergensky	0.49	1.65	4.35	6.85	11.00	15.4	20.5	27.00	34.0	40.8	48.5	56.5			
	Darcy	0.46	1.85	4.2	7.4	11.6	16.7	22.74	29.7	37.6	46.4	56.1	66.8			
	Williams & Hazen	0.54	1.95	4.15	7.05	10.66	14.98	19.87	24.77	31.68	38.45	45.72	53.42			
	Average	0.48	1.75	3.9	6.57	10.18	14.41	19.33	24.92	31.43	38.3	45.86	54.0			
	Fanning	0.414	1.47	3.02	5.14	7.83	11.03	14.72	18.93	23.58	28.8	34.53	40.6			
	Meier	0.49	1.8	3.75	6.5	9.8	13.8	18.4	23.6	29.4	35.65	42.67	50.2			
	Harrison	....	1.2	2.72	4.54	6.82	9.56	12.72	16.3	20.2	24.8	29.8	35.2			
	Serginsky	0.38	1.53	3.44	5.75	8.55	12.00	16.4	21.00	26.5	32.0	38.00	45.0			
1½	Darcy	0.37	1.5	3.67	5.98	9.35	13.4	18.3	23.94	30.3	37.4	45.25	53.96			
	Williams & Hazen	0.45	1.63	3.46	5.89	8.91	12.5	16.56	21.31	26.82	32.18	38.38	45.0			
	Average	0.42	1.52	3.34	5.63	8.54	12.05	16.2	20.85	26.13	31.8	38.1	45.0			
	Fanning	0.286	1.06	2.25	3.88	5.94	8.33	11.17	14.3	17.87	21.7	26.04	30.7			
	Meier	0.36	1.31	2.75	4.75	7.2	10.12	13.5	17.3	21.5	26.2	31.23	36.72			
	Harrison	0.23	0.92	2.04	3.41	5.12	7.76	9.54	12.2	15.2	18.6	22.3	26.4			
	Serginsky	0.34	1.12	2.34	3.9	5.8	8.3	11.00	14.2	18.3	22.0	26.0	31.0			
2	Darcy	0.27	1.07	2.41	4.29	6.7	9.64	13.13	17.15	21.7	26.8	32.4	38.6			
	Williams & Hazen	0.34	1.22	2.6	4.43	6.7	9.36	12.48	15.98	19.87	24.12	28.66	33.7			
	Saph., galv. iron	0.226	0.865	1.83	3.12	4.71	6.6	8.9	11.3	14.1	17.0	20.4	24.0			
	Schroder, old pipe	0.315	1.24	2.77	4.9	7.63	10.9	14.8	19.3	24.06	29.9	36.1	42.76			
	Average	0.3	1.1	2.37	4.08	6.23	8.88	11.84	15.22	19.08	23.3	28.0	33.0			
	Fanning	0.232	0.861	1.86	3.2	4.9	6.88	9.18	11.84	14.8	18.0	21.7	24.8			
	Meier	0.28	1.05	2.25	3.8	5.75	8.44	10.8	13.9	17.26	21.0	25.1	29.5			
	Harrison	0.24	0.72	1.63	2.72	4.1	5.72	7.62	9.8	12.22	14.92	17.9	21.0			
	Serginsky	0.255	0.795	1.7	2.89	4.35	6.2	8.2	10.9	13.6	16.5	19.8	24.0			
2½	Darcy	0.21	0.85	1.91	3.39	5.31	7.65	10.41	13.6	17.21	21.25	25.71	30.6			
	Williams & Hazen	0.27	0.98	2.1	3.57	5.39	7.56	10.05	12.89	16.03	19.5	23.22	27.25			
	Average	0.24	0.88	1.91	3.26	4.98	7.07	9.38	12.16	15.2	18.53	22.24	26.2			
	Fanning	0.18	0.68	1.47	2.52	3.78	5.47	7.29	9.37	11.7	14.27	17.2	20.4			
	Meier	0.22	0.8	1.7	2.85	4.4	6.2	8.92	10.6	13.14	16.0	19.1	22.45			
	Harrison	0.151	0.604	1.36	2.28	3.41	4.78	6.36	8.16	10.1	12.4	14.9	17.2			
	Serginsky	0.202	0.645	1.35	2.32	3.5	5.0	6.7	8.8	10.9	13.2	15.7	19.0			
3	Darcy	0.161	0.644	1.45	2.58	4.03	5.8	7.9	10.3	13.05	16.11	19.5	23.2			
	Williams & Hazen	0.216	0.77	1.63	2.79	4.22	5.9	7.85	10.12	12.54	15.21	18.4	21.28			
	Saph.& Schroder,w.i. pipe	0.161	0.609	1.32	2.29	3.5	4.96	6.7	8.6	10.7	13.2	15.8	19.0			
	Average	0.18	0.68	1.47	2.52	3.84	5.45	7.4	9.42	11.73	14.34	17.2	20.35			
	Fanning	0.153	0.58	1.25	2.15	3.29	4.64	6.21	8.0	9.97	12.15	14.6	17.2			
	Meier	0.18	0.66	1.42	2.45	3.65	5.2	7.0	9.0	10.9	13.2	15.9	18.6			
	Harrison	0.13	0.52	1.17	1.95	2.94	4.07	5.42	7.0	8.72	10.65	12.8	15.1			
	Serginsky	0.165	0.53	1.12	1.9	2.84	4.1	5.5	7.2	8.8	10.7	13.0	15.5			
3½	Darcy	0.134	0.535	1.2	2.14	3.34	4.81	6.55	8.56	10.83	13.37	16.18	19.25			
	Williams & Hazen	0.18	0.643	1.36	2.32	3.5	4.87	6.42	8.36	10.4	12.62	15.05	17.7			
	Average	0.17	0.58	1.25	2.15	3.26	4.62	6.20	8.02	9.93	12.12	14.6	17.2			
	Fanning	0.132	0.503	1.09	1.87	2.87	4.04	5.41	7.0	8.68	10.6	12.75	15.0	17.46	20.1	22.9
	Meier	0.157	0.565	1.23	2.04	3.15	4.4	5.9	7.4	9.7	11.37	13.6	16.0	18.52	21.26	24.2
	Harrison	0.113	0.45	1.02	1.77	2.56	3.58	4.77	6.12	7.64	9.38	11.1	13.2	15.3	17.5	19.86
	Serginsky	0.133	0.45	0.955	1.6	2.4	3.4	4.65	6.0	7.4	9.1	10.9	13.2	15.2	17.64	20.25
	Darcy	0.115	0.46	1.04	1.85	2.88	4.15	5.65	7.38	9.34	11.53	13.96	16.61	19.6	22.6	25.95
4	Williams & Hazen	0.155	0.55	1.19	2.02	3.06	4.29	5.71	7.66	9.09	11.04	13.16	15.4	17.86	20.45	23.18
	Saph. Schroder	0.118	0.46	1.01	1.76	2.72	3.88	5.24	6.8	8.55	10.5	12.7	15.0	17.46	20.1	22.9
	Average	0.132	0.5	1.22	1.83	2.8	3.96	5.33	6.91	8.62	10.5	12.6	14.92	17.34	19.52	22.75
	Fanning	0.103	0.39	0.85	1.47	2.25	3.16	4.23	5.45	6.8	8.3	9.94	11.7	13.6	15.6	17.7
	Meier	0.118	0.425	0.91	1.6	2.35	3.35	4.4	5.75	7.0	8.6	10.2	12.0	13.96	16.0	18.23
	Harrison	0.097	0.39	0.816	1.36	2.05	2.86	3.81	4.9	6.11	7.46	8.95	10.5	12.3	14.15	16.11
	Serginsky	0.102	0.335	0.7	1.18	1.8	2.52	3.48	4.4	5.5	6.85	8.15	9.8	11.45	13.23	15.13
	Darcy	0.088	0.35	0.79	1.41	2.21	3.18	4.33	5.65	7.15	8.83	10.7	12.7	14.9	17.3	19.9
5	Williams & Hazen	0.12	0.43	0.909	1.55	2.34	3.27	4.36	5.58	7.07	8.42	10.05	11.8	13.68	15.62	17.9
	Saph. Schroder	0.099	0.37	0.807	1.4	2.14	3.03	4.10	5.25	6.58	8.05	9.7	11.4	13.3	15.3	17.4
	Average	0.13	0.39	0.83	1.42	2.16	3.05	4.10	5.28	6.6	8.07	9.67	11.4	13.31	15.31	17.5
	Fanning	0.084		0.69	1.2	1.84	2.58	3.46	4.5	5.54	6.77	8.12	9.58	11.06	12.82	14.76
	Meier	0.08		0.73	1.25	1.9	2.63	3.5	4.5	5.6	6.9	8.5	9.6	11.09	12.73	14.5
	Harrison	0.062		0.68	1.13	1.7	2.38	3.18	4.08	5.09	6.21	7.45	8.8	10.44	12.11	13.9
	Serginsky	0.06		0.552	0.945	1.43	2.01	2.76	3.5	4.4	5.45	6.5	7.7	9.0	10.4	11.84
	Darcy	0.071		0.643	1.14	1.79	2.75	3.5	4.57	5.79	7.1	8.64	10.3	12.07	14.0	16.07
6	Williams & Hazen	0.096		0.73	1.25	1.9	2.64	3.53	4.51	5.62	6.84	8.1	9.44	10.94	12.56	14.36
	Saph. Schroder	0.067		0.506	0.88	1.36	1.94	2.62	3.39	4.26	5.23	6.3	7.5	8.74	10.09	11.52
	Darcy experiment	0.77		0.76	1.13	1.7	2.44	3.33	4.35	5.5	6.79	8.18	9.74	11.43	13.26	15.12
	Average	0.074		0.66	1.12	1.7	2.42	3.23	4.18	5.23	6.41	7.72	9.08	10.6	12.25	14.01
	Fanning	0.065		0.586	1.02	1.55	2.19	2.93	3.78	4.71	5.74	6.9	8.16	9.52	11.0	12.56
	Meier	0.066		0.6	1.02	1.57	2.17	2.9	3.75	4.7	5.7	6.9	8.0	9.23	10.6	12.04
	Harrison	0.053		0.481	0.85	1.33	1.92	2.62	3.42	4.33	5.34	6.5	7.7	9.02	10.46	12.00
7	Serginsky	0.051		0.46	0.78	1.28	1.67	2.28	2.89	3.6	4.4	5.25	6.2	7.2	8.3	9.45
	Darcy	0.06		0.54	0.97	1.51	2.17	2.96	3.87	4.89	6.04	7.31	8.7	10.21	11.84	13.6
	Williams & Hazen	0.081		0.62	1.05	1.59	2.22	2.96	3.8	4.71	5.72	6.83	8.02	9.3	10.67	12.12
	Average	0.063		0.55	0.95	1.47	2.06	2.78	3.6	4.5	5.5	6.62	7.8	9.08	10.48	11.96
	Fanning	0.056		0.505	0.875	1.34	1.9	2.53	3.26	4.07	4.96	6.0	7.11	8.22	9.5	10.84
	Meier	0.058		0.52	0.875	1.34	1.89	2.5	3.2	4.0	4.8	5.8	6.9	7.86	9.02	10.26
	Harrison	0.057		0.51	0.854	1.28	1.79	2.38	3.06	3.82	4.66	5.59	6.6	7.91	9.2	10.6
8	Serginsky	0.043		0.39	0.657	1.01	1.42	1.93	2.45	3.02	3.77	4.5	5.3	6.2	7.17	8.21
	Darcy	0.0524		0.47	0.84	1.31	1.89	2.57	3.35	4.24	5.24	6.34	7.55	8.85	10.27	11.79
	Williams & Hazen	0.07		0.53	0.91	1.37	1.92	2.56	3.26	4.06	4.94	5.9	6.9	8.01	9.19	10.44
	Average	0.56		0.5	0.835	1.27	1.81	2.41	3.1	3.87	4.73	5.7	6.73	7.84	9.06	10.36
	Fanning	0.043		0.393	0.68	1.04	1.47	1.97	2.54	3.17	3.88	4.65	5.52	6.4	7.41	8.47
	Meier	0.04		0.38	0.65	1.0	1.42	1.87	2.43	3.0	3.6	4.4	5.1	5.92	6.8	7.73
	Harrison	0.047		0.408	0.683	1.02	1.43	1.9	2.45	3.05	3.73	4.47	5.28	6.15	7.25	8.32
10	Serginsky	0.034		0.293	0.5	0.77	1.07	1.45	1.85	2.3	2.8	3.3	4.0	4.73	5.5	6.3
	Darcy	0.041		0.37	0.65	1.02	1.47	2.0	2.61	3.31	4.08	4.94	5.88	6.9	8.00	9.19
	Williams & Hazen	0.054		0.41	0.698	1.06	1.48	1.97	2.52	3.14	3.81	4.51	5.33	6.15	7.06	8.02
	Average	0.043		0.376	0.64	0.97	1.4	1.86	2.4	3.0	3.65	4.4	5.2	6.0	7.0	8.0
	Fanning			0.32	0.55	0.843	1.2	1.61	2.07	2.58	3.16	3.8	4.48	5.24	6.10	6.89
	Meier			0.31	0.525	0.79	1.12	1.5	1.9	2.4	2.9	3.5	4.0	4.72	5.41	6.16
	Harrison			0.347	0.581	0.871	1.26	1.62	2.08	2.6	3.17	3.8	4.49	5.23	6.1	6.97
12	Serginsky			0.235	0.4	0.615	0.87	1.15	1.5	1.8	2.3	2.7	3.1	3.5	3.92	4.5
	Darcy			0.30	0.54	0.839	1.21	1.64	2.15	2.72	3.36	4.06	4.83	5.67	6.58	7.55
	Williams & Hazen			0.33	0.562	0.85	1.19	1.58	2.03	2.53	3.07	3.65	4.28	5.05	5.9	6.69
	Average			0.31	0.53	0.8	1.14	1.52	1.96	2.44	3.0	3.6	4.2	4.9	5.67	6.46

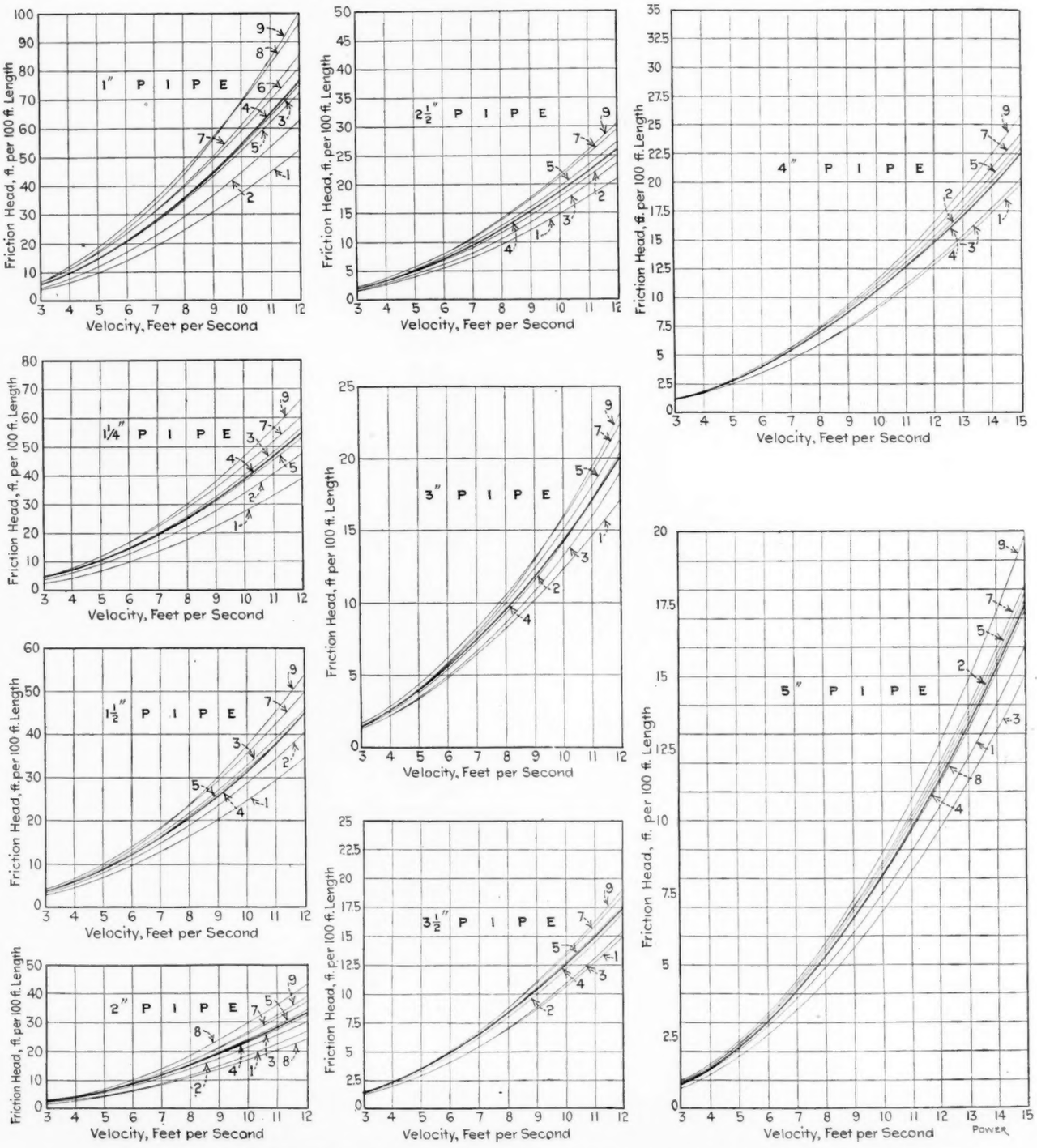
V = Velocity in feet per second;
c = Constant varying with condition of surface;
r = Hydraulic radius = $\frac{d}{4}$ in feet in this case;
h = Friction head;
l = Length of piping in feet;
$0.001^{-0.04}$ = Factor added to make numerical value of c equal to c in Chezy formula, at a slope of 0.001.

The constants are: 140 for best cast-iron pipe, tin, lead, brass, etc.; 130 for new cast-iron pipe; 120 for cast-iron pipe 5 years old; 110 for cast-iron pipe 10 years old; 100 for cast-iron pipe 17 years old.

Substituting $\frac{d}{4}$ for r and extracting the 0.54 root,

$$h = \left\{ \begin{matrix} 0.0006 \\ 0.00043 \\ 0.00037 \\ 0.00032 \end{matrix} \right\} l \frac{V^{1.85}}{d^{1.166}}$$

1 Harrison Formula, also Poché Tests for 1" Pipe
2 Fanning Formula
3 Serginsky Formula
4 Average
5 Williams & Hazen Formula
6 Poché Tests Galvanized Iron Pipe
7 Meier Formula
8 Saph Schoder Tests
9 Darcy Formula



CURVES FOR FRICTION IN PIPES PLOTTED FROM DIFFERENT FORMULAS

the numerical values in the equation depending on the respective use of the following values for c: 100, 120, 130 and 140.

In the charts and tables the factor 120 gave results for most of the sizes higher than the average. The factor 125 would give results checking very closely all the way through. The greatest variation is shown in the chart for 1-in. pipe; here the average values are high as several tests of 1-in. old and galvanized-iron pipe were included.

No. 2 is Darcy's formula which gives high values throughout. It was never intended for very small pipe sizes, say under 3 in., and was based entirely on water-works practice. The formula is expressed in several ways, but all may be reduced to the same thing. One of the simpler forms of the Darcy formula is:

$$h = 0.003732\, l\, V^2\, \frac{D+1}{D^2} \qquad (2)$$

in which

h = Friction head;

l = Length of pipe in feet;

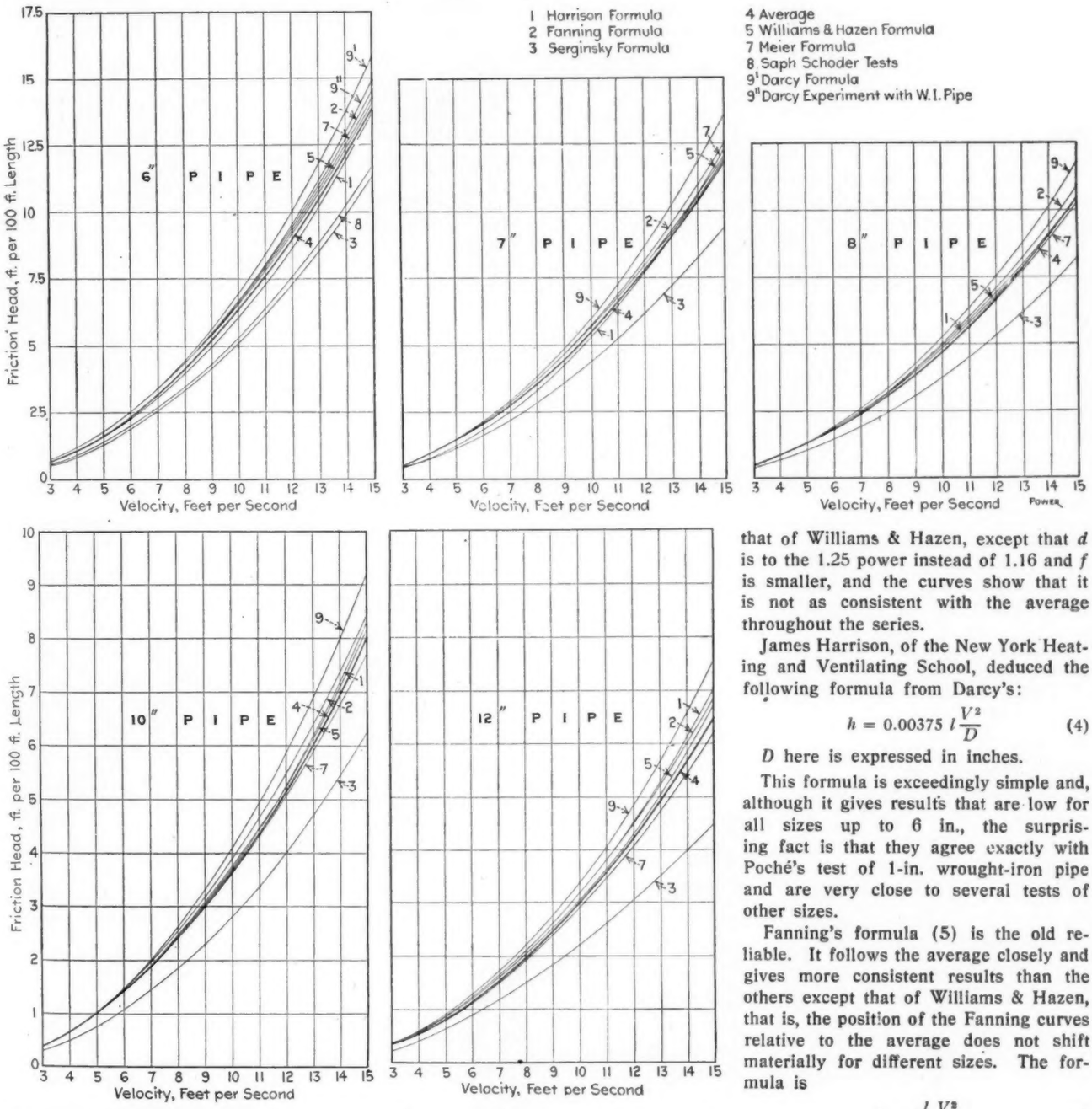
V = Velocity in feet per second;

D = Diameter in inches.

Meier's formula is

$$h = 0.0038\, l\, \frac{V^{1.86}}{D^{1.25}} \qquad (3)$$

Here D is expressed in feet. Mr. Meier increased this constant from 0.00038 to 0.0004 and introduced the factor $2g$ which complicates the formula. The increase of this friction factor or constant makes the results for all small sizes and great velocities high, and above 8-in. pipe it gives results more and more below the average. It may be noticed that this exponential formula is much the same as that of Williams & Hazen, except that d is to the 1.25 power instead of 1.16 and f is smaller, and the curves show that it is not as consistent with the average throughout the series.



CURVES FOR FRICTION IN PIPES PLOTTED FROM DIFFERENT FORMULAS (CONTINUED)

James Harrison, of the New York Heating and Ventilating School, deduced the following formula from Darcy's:

$$h = 0.00375\, l\, \frac{V^2}{D} \qquad (4)$$

D here is expressed in inches.

This formula is exceedingly simple and, although it gives results that are low for all sizes up to 6 in., the surprising fact is that they agree exactly with Poché's test of 1-in. wrought-iron pipe and are very close to several tests of other sizes.

Fanning's formula (5) is the old reliable. It follows the average closely and gives more consistent results than the others except that of Williams & Hazen, that is, the position of the Fanning curves relative to the average does not shift materially for different sizes. The formula is

$$h = f\, \frac{l}{d}\, \frac{V^2}{2g} \qquad (5)$$

Calculation with this formula is very burdensome, due to the varying values of f for each change in size and velocity. Tabulation of the final results might just as well be made as of the series of factors, and the formula would then be unnecessary. This formula is commonly used with a coefficient of 0.02 for all cases, but this is only an approximation.

For the tables and curves a complete set of factors for each size and velocity was interpolated and the latest commercial exact sizes for the diameters of the pipe.

Serginsky's formula was compiled from the latest German authorities. Although it was not available the author has confidence in the results as far as computation is concerned. These were read from a logarithmic chart and appear to be high for small sizes and low for the larger sizes. The curve changes its position with reference to the average from high in the 1-in. chart to the lowest on the 12-in. chart.

The data in this article are not in convenient shape to use in practice but charts or tables may be prepared according to any one of the formulas by plotting the results given in Tables 1 and 2. As stated before, the most consistent results are from Williams & Hazen's formula with 120 to 125 as the constant. In the use of this factor there should be a sufficient allowance for practical increase in friction over ideal conditions, but not enough to allow for careless or poor construction.

It is interesting to notice that all of these formulas involve a numerical constant, which is very close to 0.0037 in all cases.

CORRESPONDENCE

Mr. Jahnke's Proposed Coil

In Mr. Jahnke's arrangement of heating coils, shown in Fig. 2, page 842 of the June 11 issue (repeated herewith), appears at least one fault. A coil of 200 ft. of 3-in. pipe will expand considerably and expand more than the return line; therefore he will find the conditions much the same as before. He should put in more ells between the branch tees and the manifolds to allow for expansion.

W. H. H. PLOWMAN.

Philadelphia, Penn.

When Mr. Jahnke changes over in the manner he describes he will be troubled considerably on account of the steam "short-circuiting" unless he has separate drips on each of the coils.

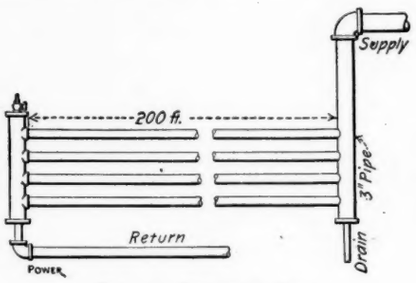


FIG. 1. PRESENT COIL

It would be better to use horizontal manifolds, as in Fig. 2, on the steam and also on the drip of the present coil, Fig. 1. By using elbows and having the drops from and to the headers about 3 or 4 ft. long to allow for expansion, he will not have any trouble with the old coil. The coil should pitch as much as convenient toward the drip, which should be of sufficient size to take care of the condensation.

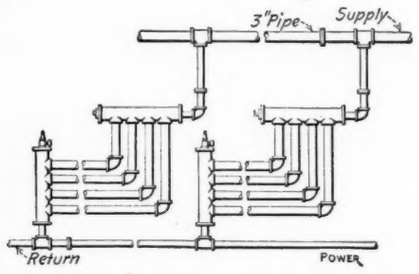


FIG. 2. PROPOSED COIL

I. M. COLEMAN.

Ipswich, Mass.

TABLE 2. CAPACITIES OF PIPES AT VARIOUS VELOCITIES

Nominal Dia., In.	Actual Dia., In.	Dia., Ft.	Area, Sq.Ft.	Cap. per Min.	Velocity Feet per Second										
					1	2	3	4	5	6	7	8	9	13	14
1	1.05	0.0875	0.0062	Cu.Ft.	0.372	0.744	1.12	1.5	1.86	2.23	2.6	3.00	3.35	4.8	5.2
				Gal.	2.783	5.6	8.33	11.11	13.9	16.7	19.5	22.24	25.00	36.16	38.9
				Lb.	23.1	46.2	69.3	92.4	115.5	138.6	161.7	185	208	300	323
1¼	1.38	0.115	0.0104	Cu.Ft.	0.624	1.25	1.87	2.5	3.12	3.7	4.4	4.99	5.62	8.11	8.74
				Gal.	4.67	9.34	14.00	18.7	23.3	28	32.7	37.4	42.03	60.71	65.4
				Lb.	38.69	77.4	116	155	193	232	271	309	348	503	542
1½	1.61	0.134	0.0141	Cu.Ft.	0.840	1.69	2.54	3.4	4.23	5.1	5.92	6.77	7.61	11	11.86
				Gal.	6.33	12.66	19	25.3	31.6	38	44	50.6	57	82.3	88.62
				Lb.	52.45	104.9	157	210	262	315	367	420	472	682	735
2	2.06	0.172	0.0236	Cu.Ft.	1.416	2.83	4.25	5.7	7.08	8.5	9.9	11.33	12.7	18.4	19.9
				Gal.	10.592	21.2	31.8	42 4	53	63.6	74.1	85	95.3	137.7	148.3
				Lb.	87.79	175.6	263.4	351	439	527	615	702	790	1141	1230
2½	2.46	0.205	0.0332	Cu.Ft.	1.992	3.98	6	8	10	12	14	16	18	26	27.9
				Gal.	14.9	29.8	44.7	60	74.5	89.4	104.3	119.2	134	194	209
				Lb.	123.5	247	371	494	617.5	741	865	988	1111 .	1606	1730
3	3.06	0.255	0.0513	Cu.Ft.	3.078	6.16	9.24	12.3	15.4	18.5	21.55	24.6	27.7	40	43.1
				Gal.	23.023	46	69.1	92.1	115.1	138.2	161.2	184.2	207.2	299.3	322.3
				Lb.	190.84	382	572	763	954	1145	1336	1527	1717	2480	2672
3½	3.56	0.297	0.0687	Cu.Ft.	4.122	8.2	12.4	16.5	20.6	24.8	28.9	33.0	37.1	53.60	57.70
				Gal.	30.8	61.6	92.4	123	154	185	205.7	246	277.2	400.4	431.2
				Lb.	255.6	511	767	1022	1278	1534	1789	2044	2301	3323	3578
4	4.02	0.335	0.0884	Cu.Ft.	5.304	10.61	15.91	21.22	26.52	31.8	37.1	42.44	47.7	68.95	74.25
				Gal.	39.674	79.4	11.9	159	198	238	278	318	357	516	555
				Lb.	328.85	658	987	1316	1645	1973	2303	2631	2960	4275	4604
5	5.04	0.42	0.139	Cu.Ft.	8.34	16.7	25	33.4	41.70	50	58.4	66.7	75	108.4	116.8
				Gal.	62.38	124.76	187.	250	312	374	437	500	561	811	874
				Lb.	517.1	1034	1551	2068	3586	3102	3620	4137	4654	6722	7239
6	6.06	0.505	0.201	Cu.Ft.	12.06	24.1	36.2	48.2	60.3	72.4	84.4	96.4	108.6	156.8	169
				Gal.	90.21	180	271	361	451	542	632	722	812	1173	1263
				Lb.	747.7	1496	2243	2991	3739	4486	5234	5982	6729	9720	10,468
7	7.02	0.585	0.269	Cu.Ft.	16.14	32.3	48.4	64.6	80.7	96.8	113	129.2	145.2	209.8	226
				Gal.	120.73	242	362	483	604	725	846	967	1088	1570	1691
				Lb.	1000.7	2001	3002	4,003	5,004	6,004	7,005	8,006	9,006	13,009	14,010
8	7.97	0.665	0.347	Cu.Ft.	20.82	41.6	62.5	83.3	104.1	125	145.8	166.6	187.5	270.7	291.5
				Gal.	155.74	311.5	467.2	623	779	935	1,090	1,246	1,402	2,025	2,180
				Lb.	1290.8	2582	3872	5,163	6,454	7,744	9,035	10,326	11,616	16,783	18,073
10	10.01	0.834	0.55	Cu.Ft.	33	66	99	132	165	198	231	264	297	429	462
				Gal.	247	494	741	988	1,235	1,482	1,729	1,976	2,223	3,211	3,458
				Lb.	2046	4092	6138	8,184	10,230	12,276	14,322	16,368	18,414	26,580	28,644
12	1 .2	1	0.787	Cu.Ft.	47.22	94.44	141.66	188.9	236.1	283	330	378	425	614	661
				Gal.	353.21	706.4	1059.6	1,421.8	1,766	2,119	2,472	2,825	3,179	45,916	4,945
				Lb.	2928	5856	8784	11,712	14,640	17,568	20,496	23,424	26,352	38,060	40,990

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POWER

The Turbine Contest

We trust that the participants in the turbine contest are well endowed with patience. The articles submitted have passed through the hands of two of the judges and by the time this paper reaches the subscribers will be in the hands of the third and last. The judges are all busy and important men, and we are sure that the contestants, as well as the editors, appreciate their kindness in consenting to serve. The conscientious reading and comparing of the merits of the large number of manuscripts submitted involves much time and labor. There shall be no unnecessary delay in arriving at a decision, and it will be announced and the prize awarded, as soon as the judges have spoken.

Unreliable Data

In the office of a prominent consulting engineer, a draftsman, laying out the heating mains for a large building, asked how he should determine the pipe sizes. The consulting engineer brought out a blueprint of tabular results supposed to give the proper size for different lengths and capacities for one pound drop in pressure. Asked if he was sure of the data, he replied: "I guess the figures are all right as they are used by a large company engaged in this line of work." It so happened the table did not apply, as it was based upon altogether different conditions. Such procedure is quite common and has resulted in the design of many unsatisfactory, or at least inefficient systems.

There is hardly a branch of engineering where more actual guessing is employed, than in the sizing of conduits for fluids, as applied to the practice of heating. Even when tables are employed the user seldom knows their origin or limitations and as infrequently checks them carefully. If the results have failed to give trouble through a period of use it is sufficient.

Actual tests on friction of fluids in commercial pipes are comparatively few in number and meager in detail. And in most cases the tests relate to waterworks practice where the kind of pipe and surface differs from that for heating. What tests there are give a wide variation of results.

Most of the friction formulas are fairly accurate if the correct constant for roughness can be assumed or determined, but calculating a series of results for final sizes and velocities is so laborious that ordinarily pipe sizes for steam and water heating are guessed at, which may be as good as guessing at a coefficient for roughness.

Some interesting data on this subject appear on page 54 of this issue. The article compares in detail the results obtained from several standard formulas for the loss of head in friction due to the flow of water in standard commercial wrought-iron pipe. The results are tabulated and also plotted in curves, and an average in both cases has been determined.

Many authorities no doubt have investigated the subject in much the same way but the results of their investigations appear only in their formulas. The present article is arranged so that the reader can make his own comparisons and form his own conclusions. He can also vary the constant of any formula and judge of the effect on the general result. The tables and curves have been arranged more for purposes of comparison than for easy practical use, but the data can be rearranged easily to suit individual conditions. It is to be hoped that the matter will prove as valuable as it was tedious and laborious for the author to compile it.

The purpose of the investigation was to determine the formula and constant best suited for designing hot-water heating systems and in this connection it would be interesting if readers of POWER in possession of special formulas or tests on commercial wrought-iron pipe would send them in for comparison with the data presented in the article.

Ash and Clinkers

"Little satisfactory information regarding the impurities found in coal, their nature and appearance, action in the furnace and effects on combustion, is to be found in textbooks, or technical literature generally. When the subject is mentioned it is usually dismissed in a few lines, or, at most, in two or three short paragraphs, which discuss merely the effect that ash in general has upon fuel economy and combustion, but do not go into the question of the nature of the ash, or describe the effect which its different constituents have in reducing the efficiency of the coal, and in causing the formation of clinkers," says a writer in *Coal Age*.

This is all too true and, moreover, perhaps due to the paucity of literature on the subject, it is too superficially considered by steam users. The greatest number of heat units per ton at the lowest price is desired in a coal. If it is high in ash it is generally worth less per unit of weight and is far more expensive to handle. But because it is low in ash does not signify that it is really the cheapest coal for the purpose. The nature of the ash may be such that when the bed of ash on the grate reaches the fusing temperature, the most troublesome kind of clinkers will form, especially if the coal contains much sulphur, lime, iron oxide, silica and other impurities which tend to lower the fusing point of the ash.

Ash is not only a total loss when the thermal value of a fuel is considered, but a great amount of heat in the coal is wasted in raising the temperature of the ash to that of the fuel bed. Additional expense is due to the labor necessary to handle it, and the rapid deterioration of the furnace walls and grate caused by the chemical impurities of the coal and accelerated by the frequent use of the slice and tamping bars.

When present in large quantities in the furnace, ash necessitates that the fire-doors be open for a considerable time while removing it. This allows large volumes of air to enter, seriously lowering the CO_2 and conducing to undesirable stresses in the boiler due to extreme temperature variations in its structure.

On the other hand, if the ash readily fuses, it clogs the air spaces of the grate, causing imperfect combustion, smoke and further thermal losses. Some coals form an ash which readily fuses and becomes heavy, black and ropy when hot. Its viscosity is such that it runs over the grate, filling the air spaces and finally crystallizing when cold. The most skillful fireman cannot get good results from such coal, however high it may be in heat value. It is also next to impossible to effectually clean grates once subjected to this treatment.

There is much in this subject of the impurities and clinker-forming properties of ash, and it is to be hoped that the Bureau of Mines and the various engineering colleges and societies will take it up and give publicity to the results of their investigations.

Learning by Doing

The article appearing on another page, "Preparing for License Examination," contains many ideas pertinent to the welfare of steam engineers who would progress in their work.

There are two ways of learning, by study and by experience, but both used together make a strong combination and a power for advancement.

In the case under consideration, study and experience were combined. After reading how to do a job the actual work was performed, and the idea firmly fixed in the mind.

The engineer used problems found about his plant, figured them out, and when he got through, knew a lot more than before the study period began.

Instructions for setting the valves on a slide-valve engine are published in most handbooks dealing with steam engines, but a beginner could read them several times and not obtain so complete an understanding of how the work should be done as would be obtained by performing the operation with an engine, using the printed explanation for a guide.

Other power-plant performances can be worked out in a similar manner. The idea is sound and practical. It can be adopted with profit by those who aim to accomplish things in the steam-engineering field.

On May 7 a conference was held between the Board of Public Utility Commissioners of the State of New Jersey and representatives of nine railroads operating in that state. A memorandum issued by the commission states that "The representatives present were unanimously of the opinion that there was no mechanical device that would prevent the dropping of crown-sheets in locomotive boilers, thereby preventing explosions."

On June 20 the Jacobs-Shupert U. S. Firebox Co. let the water in a locomotive boiler with one of their fireboxes go out of sight and kept up a roaring oil fire in the box until the pressure went down from 230 pounds to not enough to blow the fire without any outward manifestation of distress. A radial-stay boiler subjected to the same treatment dropped its crown-sheet with a great outpouring of steam and flying of hot firebrick and rivet heads when the water line had fallen 14.5 inches below the crown-sheet.

Is it time to call another conference?

The straight-flow steam engine, brought into prominence in recent years, through the work of Prof. Stumpf, of Charlottenburg, which has produced such astonishing results in the use of steam, was patented in England by Todd, in 1886, his patent bearing the number 2136. The Willans engine, which was constructed upon somewhat different lines, involves the same principle.

It is said that the 25,000-kw. Parsons turbine-driven unit recently purchased by the Commonwealth Edison Co., of Chicago, will be run by four boilers, each having 12,000 sq.ft. of heating surface and 250 sq.ft. of grate. This is a ratio of heating to grate surface of 48 to 1, and 1.92 sq.ft. of heating surface and $^1/_{25}$ of a square foot of grate surface per kilowatt.

A card in a Boston electric power house, says the Boston *Transcript*, reads: "To touch these wires means instant death! Anyone failing to respect this warning will be prosecuted and fined." It would be decidedly interesting to know the verdict rendered on the first case successfully prosecuted. We might even be willing to pay the fine.

At the end of fifty-seven years of activity, the Manchester Steam Users' Association is able to boast that no life has ever been lost by the explosion of any boiler under its care.

New Analysis of Indicator Diagrams

Report of the discussion upon J. Paul Clayton's paper before the A. S. M. E. entitled "A New Analysis of the Cylinder Performance of Reciprocating Engines."

This analysis, as explained in POWER, June 18, page 858, is made by transposing an ordinary indicator diagram to logarithmic coordinates.

DISCUSSION

In discussing J. Paul Clayton's paper on the above subject at the recent meeting of the American Society of Mechanical Engineers, Arthur L. Rice, of the *Practical Engineer*, said that he had had occasion to apply the same method of analysis to some diagrams taken upon tests of engines which were of only one hour's duration and found a difference of from 2 to 6 per cent. between the water rate as indicated by the analysis and that found by observation.

Prof. F. E. Cardullo, of the New Hampshire College, said that the law $pv^n = c$ is no doubt a very close approximation to the truth in the case of gas engines and air compression, although it probably is not exactly true. In the case of the adiabatic expansion of steam, if the relation of the pressure and volume is plotted on logarithmic paper the expansion curve is convex toward the origin but the curvature is very slight.

If, instead of assuming that n is a constant it is assumed that n is a variable and that the range of variation during any given expansion is very slight and nearly uniform, we shall have a still more exact statement of the case and it is unquestionable that any appreciable irregularity in the logarithmic expansion or compression line is *prima facie* evidence of leakage or of some other disturbing condition.

The author's statement that the skin surface of the cylinder must be heated once every cycle from the temperature acquired by contact with the exhaust steam nearly up to the temperature of the admission steam is misleading, since the range in temperature of the metallic surface is known to be comparatively small, being only from a fifth to a tenth of the range of the temperature of the steam.

From the author's statement it is to be inferred that the phenomena occurring after cutoff are independent of the form of the cylinder and the relation between the volume and the clearance surface. Professor Cardullo believes that a further study will lead to a change of view in this regard. He did not believe that the method is sufficiently accurate to warrant its use in determining steam consumption, although it will probably be very useful in determining losses from leakage and in showing whether or not an engine is operating under the most economical conditions possible.

He asked how the method would check itself when the steam consumption of the high-pressure and low-pressure cylinders of a compound engine is determined independently by its use. Does it give the same steam consumption for both cylinders as it should? The method is not applicable when the load on the engine is varying, since the temperature conditions of the cylinder walls are those acquired from a previous revolution and the quality at cutoff and the form of the expansion line will be different from what it would be were the engine operating with the same cutoff under a steady load. He offered what he referred to as a "more elegant, more accurate and more simple method for determining the clearance than that proposed," a description of which will be found upon pages 33 to 35 in Professor Cardullo's "Practical Thermodynamics," and will appear in the next issue of POWER.

There is no question but what the logarithmic analysis will show evidence of leakage in most cases. In certain cases, however, when there are simultaneous leaks of inlet and exhaust valves, it is quite possible that no leakage can be detected. There is also the case where the leakage takes place by the bodily transfer of water under the surface of a slide valve, which is the form of leakage most common in high-speed engines. In his opinion there is no possibility of approximating the amount of leakage from the logarithmic diagram. Data in regard to the relation of the quality of the steam at the cutoff and the exponent of expansion obtained from forms of valves known to be subject to leakage should be regarded with suspicion.

If experience justifies the claims of this form of analysis we may shortly expect to see indicators on the market which will draw the logarithmic diagrams directly upon the indicator card. The mathematical relations required are quite simple and there ought to be no difficulty in constructing an accurate and simple apparatus for the purpose. Of course, such an indicator will not draw correct diagrams unless the clearance of the engine has been determined and the apparatus adjusted accordingly. If, however, the apparatus can be adjusted until the expansion lines are straight, it will be possible to use the apparatus without determining the clearance. With regard to hooked diagrams, Professor Cardullo said that he had found a hook due to cylinder condensation in the compression curve of a diagram from a 7x12-in. Straight Line engine running at 250 revolutions per minute. These hooks will always occur with any type of engine when the compression is sufficiently great, is more pronounced in engines having small clearance volumes, and the higher the rotative speed the higher the compression pressure at which they occur.

R. C. Stevens, of the Skinner Engine Co., called attention to the failure of the method to give any account of the steam which leaked from the steam chest through the exhaust side of a piston or slide valve, without ever going into the cylinder. He called attention to the large amount of such leakage shown in the paper presented by Messrs. Dean and Wood to the society in June, 1908, entitled "Economy Tests of High-Speed Engines," and to the large amount of leakage shown by the experiments of George Mitchell and reported in POWER of Oct. 11, 1910, in an article entitled "Leakage Through a Piston Valve," in which the leakage in some cases ran as high as 22 per cent. of the total consumption. Further, in POWER for Nov. 8 of the same year, A. L. Ide & Sons state that the steam consumption of an engine was increased from 26 to 32.1 lb. per indicated horsepower-hour by a valve ten-thousandths of an inch under size, as compared with one having ordinary commercial fit. This represents a leakage of 23 per cent.

What is true of piston valves is also true of the flat-balanced valve with the pressure plate. Tests on one such engine of 80 hp., which the owners considered in good condition, showed 624 lb. of steam leakage per hour, or about 31 hp. wasted, or over 25 per cent. of leakage. It is common practice to allow 3 to 4 thousandths of an inch freedom between the valve and pressure plates when newly fitted, as in operation the valve wears small and the seat wears large. The experiments of Callendar and Nicolson in Canada as early as 1897 (Institution of Civil Engineers, *Proceedings*, 1897-1898, Vol. CXXXI) called attention to the importance of valve leakage as affecting economy and to the fact that the leakage from the steam chest directly into the exhaust is very considerable when the engine is in operation and much more at this time than when the engine is at rest. He had proved that the substitution of an expanding valve which is and remains steam-tight, produces a saving of 20 per cent. or more as compared with the usual plug valve, whether piston or flat, and he would not except the piston valve with packing ring or the Corliss or oscillating valve as used in four-valve high-speed engines from this unfavorable compari-

son, if these engines had seen a few months' service.

He challenged, therefore, Mr. Clayton's assertion that the results of the application of his methods made up to the present, with the restricting conditions imposed, tend to show that the steam consumption of engines may be approximated from the indicator diagrams to within an average difference of less than 4 per cent. from test results, and was forced to believe that Mr. Clayton's tests were made on engines in much better condition than is usual, even with new engines, or that some of his data are unreliable. For, if 20 per cent. or more of the steam that enters the throttle goes out of the exhaust without doing useful work or in any way making its presence felt on the indicator, it is difficult to understand how any analysis of the indicator diagram, or its derived logarithmic diagram, can establish the actual steam consumption of an engine to within even 20 per cent. of accuracy.

Sandford A. Moss, of the General Electric Co., said that this paper was the first new thing in its line that we have had for many years and he believed that it would be a permanent addition to steam-engine theory. Leakage must be taken into account, as Mr. Stevens has said.

Mr. Clayton's method will disclose cases where there are leakages by comparison of actual weight with figured weight upon the slope of the logarithmic lines. It seemed to him that the tests did not warrant any distinction between different pressures and that the authors should use the equation from the average of all pressures, rather than the chart presented. (See POWER, June 18, p. 863.)

Prof. William G. Ennis, of the Brooklyn Polytechnic Institute, said that having been in possession of an abstract of this paper some weeks before its appearance in the *Journal*, he had been able in one or two instances to verify Mr. Clayton's law. A small automatic engine gave $n = 0.96$ with a dryness factor at cutoff of 0.62. A Corliss engine gave $n = 1.02$ average, corresponding with an indicated steam rate of 26 lb. per i.hp.-hr. noncondensing. Some of the indicated tests in Mr. Barrus' book on "Engine Tests" give values of n which seem to be out of harmony with the Clayton law.

We may hope that the accuracy of the method will be firmly established and that a satisfactory expression may be found for the exponent of the expansion curve under other conditions, so that this method of testing may not be limited permanently to the case of engines running noncondensing. Should this hope be realized the general average of indicator practice will have to be greatly improved.

A study of values of n for the compression curve is interesting. Values exceeding 1 have frequently been found along with such absence of high final temperatures as demonstrates that the customary Hirn assumption of initial dryness equal 1 must be incorrect. He had found in two cases with small automatic engines n for compression equal to 1.85 and 1.80.

From the thermodynamic side the paper is of particular importance from the new light it throws on the question of heat interchange. We have been taught in the books that during the early part of expansion the steam continues to give up heat to the cylinder walls; afterward, as its temperature falls, again receiving the heat. But, if the expansion line of the steam diagram conforms to the law, $pv^n = con.$ (and he had before him 18 transferred diagrams in which it does), there can be no such reverse transfer. The steam must be either steadily gaining or steadily losing heat, and, according to Mr. Clayton, it loses or gains according as it is dry or wet at the point of cutoff.

For some time past he had been trying to find a case where this reversal of transfer occurred; that is, where the expansion line recrossed the adiabatic. The only places where this seemed to happen are in the books; and in the light of the present paper he was inclined to think that the apparent reversals have been generally due to leakage. In the Callendar and Nicolson experiments, the steam temperature dropped to that of the metal on the cylinder head in $^1/_{18}$ of a second after cutoff. Considering the lag and dampening of wall-temperature fluctuation, the condition thus favors a gain of heat by the steam practically throughout expansion. Strangely enough, also, the wall temperature taken at a point in the sidewall just inside the face of the head began to fall while the walls were about 17 deg. F. cooler than the steam.

The subject of steam leakage is one of the most important which the paper suggests. In this connection the discussion is especially complete because the Clayton analysis leads to conclusions regarding leakage in both condensing and jacketed engines. A large part of what we have been discussing as cylinder condensation for some years past has certainly been piston and valve leakage. Tests for tightness with the parts stationary give no indication of the leakage conditions in an engine when well warmed and running. We have been in the dark on the whole question not only as a quantitative but even as a qualitative matter.

Leakage between two chambers will, other factors being constant, be proportional to the pressure difference and inversely proportional to the seal of the valve. The diagram sketches these variables and shows that leakage is most likely to occur:

(a) Through the *steam* valve just after cutoff or just before admission.

(b) Through the *exhaust* valve during the early part of a stroke.

(c) Through the *piston* during the early part of a stroke.

In cases *b* and *c* the diagrams are scarcely likely to show leakage until after cutoff has occurred.

Leakage through the steam valve occurring just after cutoff may be shown by a steam-chest diagram. It raises the expansion curve during the early part of expansion. When it ceases the true polytropic curve begins.

Preadmission leakage at this point raises the compression curve and may take place at such a rate as to give a curve apparently polytropic with an extremely high n value. A steam-chest diagram may confirm the diagnosis.

Leakage through either piston or exhaust valve occurring just after cutoff depresses the expansion curve. If it is the piston that leaks, a corresponding rise of pressure may in some cases be noted on the exhaust line of the companion diagram for the other end of the cylinder.

If such leakage occurs before cutoff the diagram in question will not show it, but the compression diagram may show a rapid rise of pressure during the early part of its exhaust period.

Mr. Clayton's study, important and thoroughly presented as it is, is thus far but a beginning. The bearing of the new law on the application of Hirn's analysis and the temperature-entropy diagram for actual tests will compel us to revise our judgment as to the utility of these forms of investigation. A new and undoubtedly fruitful field has been opened to steam engineering.

S. W. Marquis, of the engineering-experimental station of the University of Illinois, spoke of the advantages offered by the method in testing locomotives. The paper contained but one locomotive test, but Mr. Clayton was making a thorough study of the application of his methods to locomotive engines and finding a slightly different relation between the exponent of the curve and the quality of the steam at cutoff than those reported in the present paper.

J. B. Stanwood, of the Houston, Stanwood & Gamble Co., called attention to the logarithmic treatment of the expansion curve in Prof. Perry's discussion of Mr. Willan's paper on "Non-Condensing Engine Valves." (See *Proceedings* of the Institute of Civil Engineers of Great Britain, Vol. XVIII, 1888.) He felt that there was need for further experimental work and that the relation between the exponent and the quality of the steam should be investigated for engines of other classes and under other conditions.

In the instance at hand, leaving leakage out of account, it would seem that the initial quality of steam at cutoff is dependent upon two things: First, the amount of steam actually condensed up to cutoff, and, second, the quality of steam in the clearance space just before the admission. If the quality is based

simply on the difference between steam as accounted for by the indicator at cut-off, and the water consumption as determined by the amount condensed in the condenser he feared that the result is not correct, as no consideration has been taken of the quality of the cushion steam. It may be possible that the quality of cushion steam may be as low as 10 per cent., especially if Mr. Willan's theory is true that water retained on the clearance surface forms an important factor in creating cylinder condensation. Professor Zeuner has stated that if he knew absolutely the quality of steam at any point during the compression he could determine the steam consumption of a steam engine from the indicator card, leakage neglected. This question of the quality of cushion steam may have something to do with the variations in the results obtained by Mr. Clayton in his comparison of different engines. There is quite a difference of opinion on this point of wet or dry cushion steam. Some engineers maintain that the steam is dry during compression; others that there is more or less water contained in the clearance space.

The writer is of the opinion that there are few, if any, engines which are operating without some leakage. In regard to the 12x30-in. engine, tested by Mr. Clayton, the speaker thought there must be quite an appreciable amount of leakage. Mr. Clayton states that this engine, with 109 lb. absolute pressure at the throttle, with 25 per cent. cutoff, has a quality of steam of 58.8 per cent., which means a cylinder condensation of 41.2 per cent., which is high for this cutoff in comparison with most tests that have been made on Corliss engines. George Barrus in his book on "Steam Engine Tests," estimates that the amount of condensation for 25 per cent. cutoff on an average is 24 per cent. If, therefore, there was no excessive leakage in this engine, the experiments show a remarkable difference between condensation in small engines and condensation in larger engines of the same general type.

This question of leakage has been very thoroughly investigated by a special committee of the Institution of Mechanical Engineers of Great Britain, Professor Capper, chairman. (See *Proceedings* for 1905, beginning with page 171.) The experiments conducted by him show the amount of leakage of the valves at rest as compared with the leakage of the valves in motion, and corrections were applied so that the leakage was differentiated from the initial condensation. It is this fact that leads the speaker to believe that the 12x30-in. engine should be tested for leakage, both through the steam-valve system and exhaust valve.

The discovery of Mr. Clayton may lead to the solution of some problems heretofore unsolved. We may be able to determine after some experience, by means of data heretofore unobtainable, the actual quality of steam in the compression spaces.

By comparison of different cylinders of the same general type, manufacturers may be able to predict the steam consumption, especially as to the influence of the size of cylinder.

We may be able to discover the effect of clearance and clearance spaces on the economy of the steam engine by varying the same on a given engine and then from the character of the expansion curve determining the effect of cylinder condensation.

Designers may be able to improve the proportion and construction of steam cylinders and valve-gears by the character of the expansion curve under different conditions analyzed by this logarithmic method.

The speaker feared that a determination of steam consumption of many types of engines by this method would not be very reliable. There are many cylinders in which the steam is short-circuited directly from the steam side of the cylinder to the exhaust side of the cylinder, and no evidence of this loss appears on the diagram, so that the use of Mr. Clayton's method would not be at all satisfactory.

Prof. R. C. H. Heck, of Rutgers College, submitted a discussion dealing particularly with the character of the heat interchange between cylinder walls and steam and drawing the conclusion that except for the extreme cases of very low speed and very short cutoffs, there was nothing in the showing to antagonize the empirical result of practically constant exponent *n*.

In summing up, Mr. Clayton said that he was fully aware of the extent of valve leakage and was at work upon that phase of the subject. Valve leakage by his method in certain cases, by accounting for what was in the cylinder, has been approximated to within 6 per cent. in cases with which he has had experience.

Standard Pipe Flanges and Fittings.

Edward B. Denny, president of the National Association of Master Steam and Hot Water Fitters, in his presidential address at the recent convention at Atlantic City, alluded to this subject as follows: "Most particularly would I call your attention to the report of the Committee on Standardization. It will describe in detail our efforts to establish "The 1912 U. S. Standard" schedule of Standard Weights and Extra Heavy Flange Fittings and Flanges.

"In view of the fact that some violent opposition to this schedule has been expressed by some of the manufacturers of such fittings, and that some misleading statements have been made relative to its adoption, it may not be amiss for me to state a few facts concerning the matter.

"There has been no contention as to the desirability and need of a universal standard for flange fittings and flanges. Neither has there come to our notice any criticism of the schedule adopted as to its intrinsic merits, its adaptation to the requirements of the work for which such fittings are used. On the contrary, we have received many letters of commendation from the very highest authorities for not only working out and adopting a schedule but for the evident care with which "the 1912 U. S. Standard" schedule has been prepared, reflecting, as it does, the theoretical demands of the engineer and the practical requirements of the practical steam fitter.

"So far as we are informed, the opposition to the acceptance of this schedule is based either upon the method of its adoption or the expense entailed for the necessary alterations of some patterns. As to the latter objection, it is a self-evident proposition that no standard schedule can ever be made and come into general use without requiring such changes in patterns. While this may seem a hardship to the manufacturer, he has known for years that it was inevitable when those whose interest demanded the reform should be sufficiently aroused to insist upon having it and had there been a reasonable response upon his side to the overtures made by us as long ago as 1905, this incidental expense would have been avoided to the extent of the cost of many new patterns made since that time. As to the method of the adoption of this schedule, if it shall be said (as it has been) that ample notice was not given to the manufacturers of our purpose to prepare and adopt such schedule, such a statement is not in accordance with the facts, as will be clearly shown by the reports of your secretary and the Committee on Standardization.

"If the right and authority of those who made and adopted the schedule be questioned, we would ask what higher authority can there be on the subject than that of the American Society of Mechanical Engineers, whose members design most of the plants in which such fittings are used and are responsible for their proper dimensions and strength, and the National Association of Master Steam and Hot Water Fitters, whose members install such fittings and are likewise responsible for loss of life and property resulting from so called standards fixed by those whose only interest consists in the profit that may be made on the sale of the goods?

"The subject is one that should receive your careful consideration in the light of all the information contained in the reports of your officers and committees and I recommend that the sentiment of our association in convention assembled be voiced with no uncertain sound."

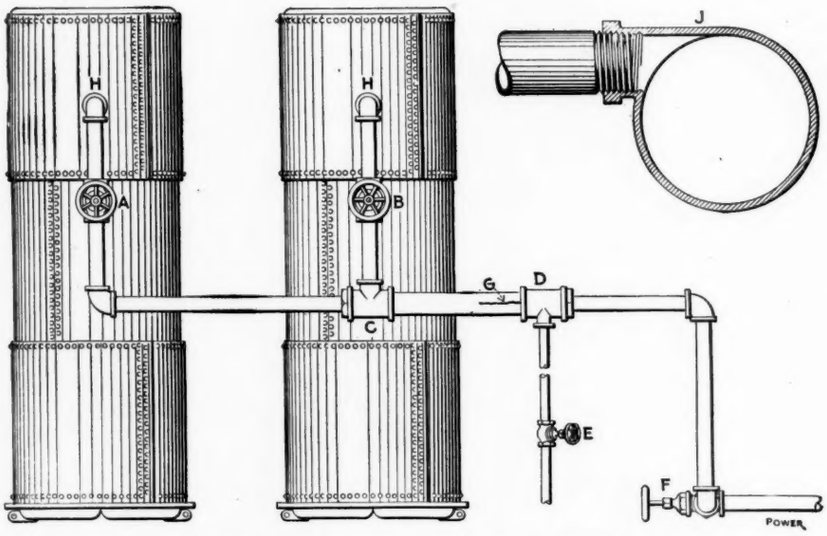
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Dangerous Steam Piping

The illustration shows the piping for a pair of tubular boilers. Each 4-in. riser is followed by an ell and a globe valve, with the stem in a vertical position, as at *A* and *B*. This is bad because it prevents water from draining automatically from the steam pipe. If globe valves are preferred, the stem should be set horizontally. From each globe valve a 4-in. pipe extends, discharging into the 4x6x4-in. tee *C* over the second boiler. This is followed by 6-in. pipe connecting with the 6x4x1-in. tee *D*.

The 2-in. horizontal pipe, which takes steam from *D*, is fitted with a globe valve *E*. Beyond this valve and nipple an ell looks downward, followed by more pipe and a throttle valve used to control a small engine.



SHOWING BROKEN PIPE OF BOILER CONNECTIONS

With a full pressure on these boilers, the valves *E* and *F* were closed for several hours; then *E* was suddenly opened to supply steam for the small engine. This was a cast-iron valve, and as soon as it was opened the whole bottom was blown off; simultaneously the 6-in. pipe burst at *G* for a distance of 10 in. When the pressure was shut off at *A* and *B* no further damage was found.

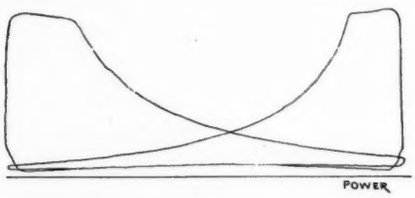
The stop valves *A* and *B* should have been replaced by angle valves, connected where the ells *H* are located. The damaged piece of pipe *G* was replaced with a 6x6x2-in. tee followed by more 6-in. pipe, a 6-in. ell and another piece of pipe carrying a 6x4-in. reducing ell looking downward. This arrangement eliminates some of the danger, as water cannot collect as before. The globe valve *E* was omitted, but a drip pipe was put in the throttle valve and a valve in it is kept open when the engine is shut down. The tee should have been of the eccentric type on the 2-in. outlet, as shown at *J*, as the pipe would have been better drained.

W. H. WAKEMAN.

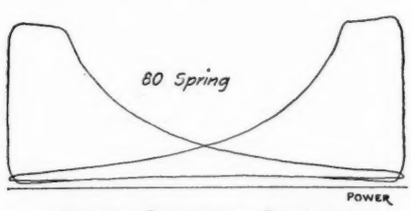
New Haven, Conn.

A Cutoff Problem

These diagrams were taken from our 20 and 40 by 36-in. cross-compound Corliss pumping engines running at 42 r.p.m., at 130 lb. steam pressure and 25.5 in. vacuum. Fig. 1 was taken after blocking the governor in running position and setting the cutoff correctly by the stroke. Fig. 2 was taken after adjusting the cutoff by the indicator diagram. The head-end steam valve cuts off the same at all times no matter at what speed the engine runs, while the crank-end valve holds on until the engine is up to speed. This fault manifests itself in both engines on both high- and low-pressure cylinders. Will some readers please tell me how to correct the fault and what causes it. The valve links on the steam valves are the same length and so are the links on the exhaust valves. The wristplate and rocker-arm swing equal distances past the center. The cutoff on the low-pressure cylinder is controlled by hand. Why cannot equal cutoff be obtained as in Fig. 2 without the head-end valve giving trouble?



SHOWING UNEQUAL CUTOFF



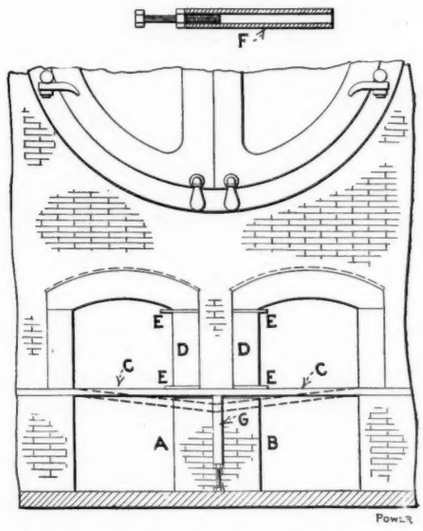
CUTOFF SOMEWHAT IMPROVED

LEROY WHITNEY.

Oakford, Penn.

Repairs to a Boiler Furnace

The piers *AB* in the illustration needed attention, but I was not willing to remove them. Examination showed that the pier *B* did not reach or support the dead plate *C*, causing the latter to bend under the weight of the upper brick work, as shown by the dotted lines, and leaving the pieces



SHOWING REPAIR TO FURNACE WITH SCREW JACK AND SHEET IRON SHIMS IN POSITION

DD loose, which at times would move out into the doorway and were likely to be broken. To remedy the trouble I cleaned all the loose material from behind the pieces *DD* and placed them in their original position, and then slipped pieces *EEEE*, of sheet iron, in the top and bottom. It was out of the question to straighten the dead plate, yet I managed to prevent it from getting worse. I then made a screw jack *F* of a 1-in. bolt and nut, and a piece of 1¼-in. pipe. This was placed at *G* on the ashpit floor under the dead plate and used to force the plate up enough to tighten the upper pier and arches.

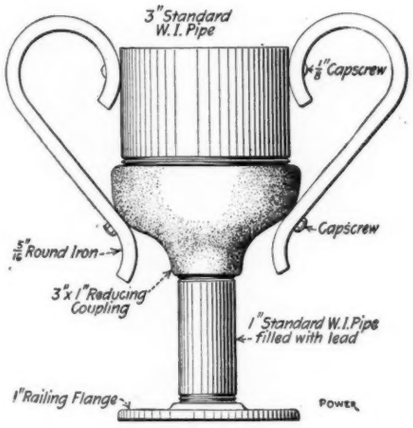
This jack and brickwork have been in place now for two years with no signs of falling down. I am surprised at the success of sheet iron as a substitute for fireclay. Many arch piers and dead plates fail on account of being inadequately supported underneath as the plate will not sustain much weight in such a hot place.

W. J. SELBY.

East Orange, N. J.

Home Made Take-down Loving Cup

This loving cup made from odds and ends found about the plant has many advantages over the common jewelry-store variety in that it is very substantial and



A UNIQUE LOVING CUP

may be easily taken apart and packed away for future use, the only tools necessary being a monkey wrench, a vise and a pair of chain tongs. This is, in my opinion, the only take-down loving cup extant.

PHILIP PARKER.

Fitchburg, Mass.

Careless Pipe Work

Fig. 1 illustrates the rear end of a tubular boiler, showing part of the blowoff pipe and connections. A reinforcing plate is riveted to the shell, to strengthen it, and a 2-in. pipe thread is cut in both plate and shell. As the combined thickness is ¾ in., and a 2-in. pipe has 11½ threads to the inch, this method gives nine threads to hold in this important but inaccessible place.

The steam fitter used a nipple that could not be screwed through the reinforcing plate; consequently it did not hold at all in the shell, as shown in the illustration. The engineer protested against this careless work, and eventually removed the nipple himself, and would not allow it to be put back. The owners then authorized the engineer to have the boiler piped according to his best judgment, which eliminated all discord between

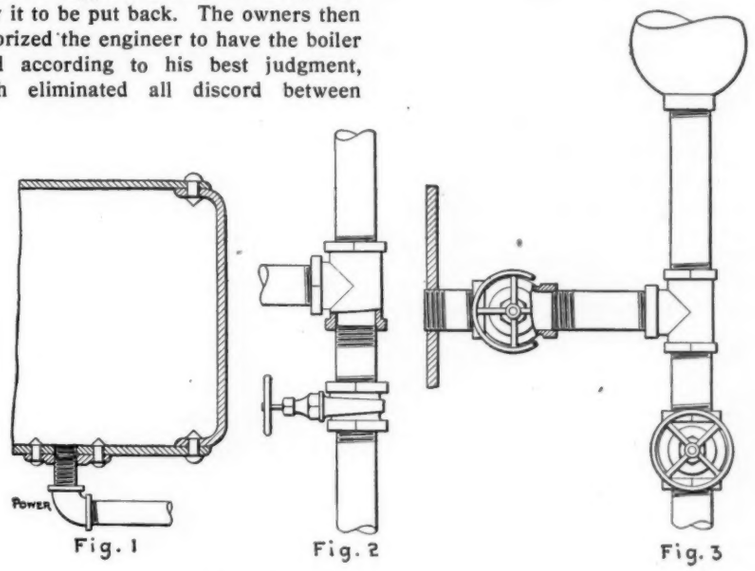


Fig. 1 Fig. 2 Fig. 3

SHOWING POOR PIPE CONNECTIONS

these two parties. The engineer had a nipple put in that extended completely through the shell but did not rise above it, thus making a flush joint that held with nine threads.

Fig. 2 shows part of a patented boiler setting consisting of 2-in. extra-heavy pipe and fittings. The gate valve shown is one of three blowoff valves required by this setting. After this boiler had been in service 14 years, the joint above the gate valve began to leak. The boiler was laid off and the water removed, and the nipple below the tee was unscrewed. When it had been given less than two turns it fell to the floor. This caused no surprise, because the nipple was supposed to be badly corroded. Closer examination showed that it was made of very thick iron pipe, and was practically free from corrosion. It had never been screwed in more than two threads, yet had held firmly and did not leak through this long term of service.

As the engineer that took it down had operated it ever since it was installed, his feelings can be better imagined than described, when he thought of the consequences if that nipple had blown out under full pressure while he was closing the valve. Such a blunder is usually made by attempting to make up both ends of the nipple at once, by putting a monkey wrench on the valve. It screws into the brass readily but is soon tight in the iron fitting; hence the failure to make two safe joints is not noticed. It should be made up with a pipe wrench in the fitting first, and the number of threads used observed; then the valve should be screwed on. A 2-in. pipe tap was screwed into the tee, and the old nipple put back, but now it holds by five threads.

Fig. 3 shows the lower connections to a water column. After it had been in service several years, it was necessary to take the column down and change its location. The heavy brass 1¼-in. nipple connected into the horizontal valve had only been screwed in less than two threads yet had held while the boiler was worked under full pressure, and its condition was only discovered when the nipple was unscrewed. Had this failed under working conditions, the result can be imagined.

W. H. WAKEMAN.

New Haven, Conn.

What Ails the Belt?

In our plant there is a 30-in. three-ply belt on one of the engines and it gives considerable trouble due to slippage. The belt is old and overloaded and more or less oil soaked. I put on a 10-in. single-ply belt as a rider and it is this rider that perplexes me. It runs with the slack side next to the tight side of the old belt and where the old belt curves down over the pulleys the rider runs as tight as a fiddle string. This engine runs "over" and the tight side of the rider is on top. I have two other belts carrying riders and they both run without trouble.

Will some interested reader state the cause of my difficulty?

A. SPENCER.

Kingston, N. C.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Lubricator Experience

The letters on lubricators have reminded me of an experience I once had with them. In the plant where I am employed are eight lubricators of various makes and sizes. When I took charge not one of them was in good condition, and the owner would not buy anything to fix them with nor get new ones, so I had to do the best I could with what material I could find.

Four of the feed-glass drain plugs were gone entirely so I took four ¼-in. pipe plugs and formed a disk on the end of each with solder. I put old globe-valve handles on what valves and filling plugs needed them and soldered seats on two of the filling plugs that would not hold tight. Not a glass had a rubber gasket on it; the glasses had been packed with asbestos wick. This was replaced with rubber gaskets. Two of the drain plugs were missing, so pet-cocks from an old pump were used instead.

The other drain plugs had been tightened with a wrench until the fiber disks had been worn, causing them to leak. I cleaned them out and ran in a mixture of lead and babbitt metal, 2: 1, and now the drain and filling plugs can be made tight by hand.

With some of the same mixture I made new disks on the try-cocks on the water column of the boiler and they have given excellent service. I put babbitt in the mixture as lead alone is too soft and wears out quickly, and babbitt alone is too hard and does not seat readily.

Since I fixed the lubricators and several other things the proprietor has not refused to get me what I needed to keep things in repair.

EDWARD AMSDEN.

Martins' Ferry, Ohio.

Failure of Boiler Tubes

In May 28 issue, Mr. Maxwell complains of failing tubes in water-tube boilers, especially at the baffle wall when it is 15 or 18 in. from the tubes.

When boiler tubes are subjected to great heat which rises above a certain intensity at any part, the walls of the tube cannot conduct all the heat to the water rapidly enough. Therefore, the outside of the tube will be hotter than the inside, which causes unequal expansion between the two surfaces and a grinding of the molecules on one another. This action prolonged for considerable periods destroys the cohesion of the particles of the tube and it fails. I had to replace the two bottom rows of tubes in four water-tube boilers on account of this action.

In each boiler these tubes had two grooves in each, extending from the front headers to the baffle bricks. These tubes had scale in the bottom half of them, and the grooves developed where the clean metal joined the portion covered by the sediment; in some places the grooves had nearly cut through the tubes and some of them developed pin holes and small leaks.

W. A. COOIL.

Ladysmith, B. C.

Corliss Engine Troubles

In the issue of May 28, E. C. Sutton, under the above heading, explains why he cannot give negative lap to the steam valves on a double-eccentric Corliss engine. If he will remove the valve-stem keys, and with the steam valve and dashpot rods in normal position, place the wristplate plumb, he can, by slipping the valve stem around in the valve crank, get any amount of negative lap he desires, but before cutting new keyways he should make sure that the valves fully close and cut off when the dashpots are down; if the valve stems are exact duplicates, he may change the head-end stem to the crank end, and the crank-end stem to the head end, and then the new keyways will come in a new place on the stems.

Unless the load on the engine requires a later cutoff, I would not recommend giving the valves negative lap, for the short lift of the dashpots on light loads is likely to cause them to give trouble. I have an engine on which the valves have negative lap and when running with light loads I have to throttle the engine.

C. P. LAMONS.

Lawton, Okla.

Mr. Sutton should place the steam wristplate at its center of travel, and remove the keys from the steam valve stems and also set the valves with the desired negative steam lap, allowing the steam valve stems to turn in the cranks without altering the length of the valve rods. Offset keys should be used to key the valve cranks in their new position on the valve stems and the steam eccentric moved back to coincide with the new position of the valves. This will give the desired negative lap and prevent the dashpot rods from striking the steam valve cranks.

Most double-eccentric Corliss engines are designed so that when the wristplate is at its center of travel the steam cranks are horizontal and a line drawn through the center of both steam cranks will cut the center of the valve stems and dashpot-rod pins.

FRED R. HAWK.

Kansas City, Mo.

Fallacy of Submerged Flotation

Neither the article on page 816 on the "Fallacy of Submerged Flotation" nor the editorial on the same subject, I believe, is explicit enough to satisfy anyone who is inclined to believe in submerged flotation, so I offer the following for such persons.

In the first place, it must be remembered that the pressure of any liquid is exerted evenly and equally in all directions, either within a receptacle containing the liquid, or on the surfaces of any object submerged in the liquid. This in itself is enough to overcome the idea which so many have that when an object sinking in the ocean reaches a certain depth the great pressure of water sustains it, for, the downward pressure being equal to the upward pressure, the object is in perfect balance so far as pressure is concerned.

Another thing to keep in mind is that any object placed in any liquid is buoyed up by a force equal to the weight of the volume of liquid displaced. This holds good whether the object floats or sinks. If an object weighs more than the liquid displaced it must sink, but if it displaces its weight of water without becoming submerged in will float.

Take a 1-ft. iron cube weighing, say, 480 lb., and assume it submerged to the depth in the Pacific Ocean mentioned in Mr. Nelson's letter, where a cubic foot of water will weigh about 67 lb. About 7.2 cu.ft. of water would have to be displaced to equal the weight of the block but as it only displaces one cubic foot of water it is evident that the buoyancy is not sufficient to keep the iron from sinking to the bottom.

If a cubic foot of iron could be cast as a hollow sphere having a volume of

7.2 cu.ft. and strong enough to withstand the enormous pressure at the depths mentioned by Mr. Nelson, it would sink only to the assumed depth of 5.9 miles, where the buoyancy would be equal to its weight.

A cubic foot of wood or anything weighing less than a cubic foot of water, if forced to any depth in the ocean and then released, would rise to the surface.

Some think a ship in sinking would be crushed to a shapeless mass while still others believe there would be numerous explosions, due to air being trapped in various compartments of the ship, the pressure of the air being raised as the pressure of the water increases.

No doubt the boilers or anything of like construction would collapse, due to the enormous external pressure, but in my opinion there would be no explosions after a ship got below the surface.

If a ship sinks bow foremost, air trapped in the stern might reach a pressure sufficient to cause an explosion before the stern gets below the water, but after the ship is completely submerged all trapped air in pockets having openings through which the water may exert its pressure will have its pressure counterbalanced by external water pressure preventing any collapse or explosion of the compartment.

Solid pieces of metal would show very little if any effect of the compression, owing to the pressure acting equally in all directions on the metal. It may be assumed also that wood or any porous substance would show no effects of the pressure as the water could saturate them, preserving the balance of pressure within and without.

As to the effect on a human body, such bones as the skull apparently would be crushed and all the bones would be crushed as greater depths were reached. The flesh being porous would show very little effect of the pressure after the water once got through the skin.

We may safely assume that submerged porous objects will not collapse or explode if not sinking too rapidly for the water to saturate them.

The question of "Submerged Flotation" is not one of pressures but of volumes and weights. Only a bed of dense silt or slime in the greater depths of the ocean could prevent an object sinking to a hard bottom.

L. A. TUCKER.

Philadelphia, Penn.

Since that great international misfortune, the sinking of the "Titanic," the question of whether a substance will float submerged at any depth in the sea has aroused considerable interest and may still be regarded an active controversy. Such an object as an ocean steamship, once beneath the surface, will sink until it meets something with greater sustaining powers than sea water. But, if this were true of all objects, fishes could not raise and lower themselves in the water with the ease they do. It is all a question of bulk for bulk and weight for weight.

An object much lighter than the same volume of water at its surface density will remain with a large part above the water. If its weight is precisely the same as that of the water it will displace, the top of the object will exactly coincide with the water surface. If now it be made a mere trifle heavier without changing its volume it will sink beneath the surface without going to the bottom. A log of water-soaked oak containing a few small bolts or spikers may be seen floating in still water with just a film above it. Another small nail will submerge it still farther. If it be forced down to such a level that the water pressure will materially reduce its volume the log will continue to sink, because its bulk is reduced by the pressure, due to the depth, much more than that of water. To float submerged in the ocean an object must be of the same specific gravity as the water at the given level and it must resist the compression due to that pressure.

The simplest object lesson which will demonstrate these principles is to submerge an egg in a tall glass vessel containing brine. Strengthen the brine by the addition of salt and the egg will approach the surface. Weaken it by pouring in fresh water and the egg will go to the bottom. In this way the egg can be held at any desired level within the limits of an ordinary sized vessel.

E. D. H. WILLARS.

Pittsburgh, Penn.

I have been much interested in the articles upon the "Fallacy of Submerged Flotation" appearing on pages 810 and 816. I would be glad if Mr. Nelson would answer the following questions:

What other great depths have been sounded in the oceans?

What are the temperatures of the water at the deep points?

What is the greatest depth from which fishes have been taken, and also the greatest depth from which other forms of life have been obtained?

How does the pressure of 13,752 lb. to the square inch at the greatest known sea depth compare with that used in hydraulic machinery?

H. S. ARNOLD.

Brooklyn, N. Y.

Mr. Tucker's statement that the question regarding an object sinking to the depths of the sea is one of relative volume and weight and not of pressure is true, but pressure tends to reduce the volume of a submerged object and also to saturate the object with water, thereby increasing its specific gravity and adding to its tendency to sink.

In regard to his suggestion concerning the hollow sphere, Mr. Tucker should remember that the difference in weight between a cubic foot of water at the surface of the sea and a cubic foot of water immediately below it is so small that the viscosity of the water as well as the friction would be sufficient to overcome the tendency to sink from one level to the next and that the object must be heavy enough to sink in the first place. If a cubic foot of water at the surface of the ocean weighs 64 lb., and a cubic foot of water at a depth of 31,614 ft. weighs 67 lb., the increased weight of a cubic foot of water per foot of depth would be $\frac{3}{31,614}$ lb., or, in round numbers, less than ¼ of a grain. This increase of weight is so small in proportion to bulk that its effect in buoying up an object is negligible. The editorial on page 810 well states the truth as follows: "Even considering a sinking body of closely the same specific gravity as water, it is inconceivable that its structure could resist the enormous hydrostatic pressure at great sea depths sufficient to prevent its specific gravity from increasing faster than that of water as it continues to sink."

Mr. Tucker is not correct in stating that after a ship is completely submerged all trapped air or air pockets having openings through which the water may exert its pressure will have a pressure equal to that on the outside of the air pocket, for the reason that the compressed air would be at an equal pressure throughout its entire bulk, whereas the pressure on the outside of the vessel would be increased about ½ lb. per square inch for each foot of depth, so that an air pocket of, say, 50 ft. in the stern of a vessel would have an interior pressure 25 lb. to the square inch greater at the top than the surrounding water. This pressure might be sufficient to throw off a deck, but such a happening would not be expected to occur, owing to the numerous deck openings, which would prevent the formation of such an air pocket, as the air would always have a tendency to force its way through any opening, no matter what pressure of water surrounds it. A vessel would at first have a tendency to sink slowly, but it would gain in velocity, and after a depth of 1500 or 2000 ft. had been reached the pressure would have increased to such an extent that boilers would collapse or their tubes rupture and the boilers fill; piping would be flattened or broken; and coal and cargo saturated. The speed of the descent would be so great that human bodies, instead of permitting saturation, due to porosity, would undoubtedly be compressed in bulk to that of a man's arm. Partially filled sealed bottles, casks, etc., would be shattered. Water at a pressure of 6000 lb., which pressure would be obtained in 2000 fathoms, would have a velocity of 1000 ft. per second; so that the smallest in-

terstices which might form pockets would be quickly penetrated and completely filled.

Mr. Willars refers to the experiment fo an egg in a glass vessel containing brine. This experiment is really useful in showing that the heaviest brine remains at the bottom and the lightest at the top. If the brine were of uniform density the egg would either float at the surface or sink to the bottom, and the fresh water referred to, which being added would cause the egg to go to the bottom, must be poured in through a tube reaching to the bottom, to produce that effect.

In reply to the questions in Mr. Arnold's letter the depth of 5269 fathoms, to which I referred, was sounded by the U. S. S. "Nero" near the island of Guam in the Western Pacific Ocean. This depth exceeds the height of the highest mountain. There are now known to be 43 areas with a depth exceeding 3000 fathoms. Depths exceeding 4000 fathoms have been found in eight different places, and two soundings have been made in depths exceeding 5000 fathoms.

The temperatures of the deeper parts of the sea are always close to the freezing point. Records of 34 deg. F. are not uncommon. Animal life which exists in the great depths is subject to enormous pressure, total darkness and intense cold.

The U. S. S. "Albatross" in the North Atlantic Ocean dredged fishes from a depth of 2949 fathoms, and it also dredged other forms of life in the South Pacific Ocean from a depth of 4173 fathoms. The same vessel also obtained 800 fishes at a single haul of the dredge from a depth of two miles in Behring Sea, which shows the abundance of animal life in great depths.

Pressures of 15,000 to 17,500 lb. per square inch are not uncommon in practical use, and special high-pressure hydrostatic apparatus have used pressures as high as 45,000 lb. per square inch.

JAMES W. NELSON.

Brooklyn, N. Y.

Carbonine

On page 266 of POWER for Feb. 20, there is an editorial headed "Get Rich Quick Schemes," the latter part of which tells about "Carbonine" and how the use of that same powder mixed with water and sprinkled on the coal is supposed to produce a saving of 25 or 26 per cent. in the coal bill. Salem is not the only city in Massachusetts where the promoters of Carbonine have caught some victims, as Fall River has two or three plants which its promoters claim are using Carbonine and for which they claim a saving of 26 per cent. in the coal pile. There are other plants which cancelled their first order for Carbonine, as I have been told, when the preposterousness of the claims of its promoters was explained to them.

It seems that the right way to use Carbonine is to buy one ton, which costs $80. Then take one pound of Carbonine powder and mix it with one gallon of water (hot or cold) it makes no difference. Then to each ton of coal sprinkle eight gallons of this mixture and then you save 26 per cent. of your coal bill. One of the promoters was asked how it was figured out that adding 67 lb. of water to a ton of coal aided the combustion of the coal, and he answered that he did not know. "You could use Carbonine without water," he said, but adding water made the mixture go farther. You could get the same results by using it dry. The writer agrees with him, and also believes he can get the same results by leaving out the Carbonine.

It seems to me that the only way to prove out this Carbonine matter, as the editorial claims, is to get some reliable mechanical engineer to make a test of the evaporative qualities of coal with and without the use of Carbonine. Then if they can add 20 or 25 per cent. to the evaporative qualities of the coal, we will believe in Carbonine, but not until then. If any man can get up a mixture that will add 20 or 26 per cent. of value to coal, then it will be, as my friend Snyder says: "We can do away with 26 per cent. of our boilers, also 26 per cent. of the boiler-making works and the Coal Barons would kidnap the discoverer."

MICHAEL H. HARRINGTON.

Fall River, Mass.

[The sample accompanying the letter below was sent without comment to Dr. C. E. Lucke, of Columbia University, for analysis, and by him submitted to a firm of industrial chemists whose report is reproduced herewith.—EDITOR.]

M. C. WHITAKER, M.Sc. F. J. METZGER, PH.D.

CONSULTING INDUSTRIAL AND ENGINEERING CHEMISTS

BROADWAY AND 116TH STREET
NEW YORK CITY

June 11, 1912.

Prof. C. E. Lucke,
Columbia University.

Dear Dr. Lucke:

We have made a qualitative analysis of the sample of material forwarded on June 5th with your letter from Power. We find that this material is practically all common salt mixed with some potassium permanganate and some zinc compound. We also note small quantities of carbonaceous material and a little sulphate.

The stuff looks to us very much like some of the fake ash burning compounds which are put out and which we have had occasion to examine before.

Hoping that you will find this satisfactory,

Very truly yours,

M. C. Whitaker

Water Broke the Piston

The pistons mentioned by A. J. Haight in the May 7 issue were undoubtedly broken by a charge of water from the steam or exhaust line. One cannot say which as the piping arrangement is unknown.

Engines that exhaust into a common exhaust header are subject to serious accidents from the presence of considerable water in the header and piping. I have seen high-speed engines wrecked the same as that mentioned by Mr. Haight and have traced the cause to faultily erected headers and piping.

C. R. MCGAHEY.

Baltimore, Md.

Wavy Expansion Line

In the May 28 issue, Leighton Johnson, in commenting on the cause of the wavy expansion line of the diagrams mentioned in the Apr. 16 issue, refers to a 1½-in. pipe forming the connection to the indicator. This is undoubtedly a draftsman error as this and the other connections are of ½-in. pipe and fittings.

J. W. DICKSON.

Memphis, Tenn.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Capacity of Stranded Cables

How is the safe current carrying capacity of a stranded copper conductor computed? From the wiring tables, a No. 12 B. & S. rubber-covered copper wire will safely carry 17 amperes. Would a cable made of 19 strands of No. 12 wire carry

$$19 \times 17 = 323 \text{ amperes}$$

or would it be determined by multiplying 6530 circ. mils (the area of a No. 12 wire) by 19, which would equal 124,070 circ. mils. The tables show a rubber-covered cable having a cross-section of 124,074 circ. mils to be capable of safely carrying only 141 amperes. Why this difference?

J. B.

The reason for 19 independent No. 12 rubber-covered copper wires being able to carry 323 amperes, whereas a stranded cable composed of the same number of wires of the same size would safely carry only 141 amperes, is that the independent wires present a greater radiating surface than the cable; hence the temperature rise is less. The combined circumferences of the independent wires is about 4.8 in. as against 1.1 in., the circumference of the cable.

The formula for figuring the current carrying capacity of single-conductor rubber-covered, stranded cable is

$$I = 490\sqrt{\frac{TD^3}{r}}$$

where

D = Diameter of cable in inches;
T = Temperature rise in cable core in degrees Centigrade;
I = Current in amperes;
r = Specific resistance of wire in ohms per inch-foot at the final temperature.

Value of Flue Gas Analysis

Is flue-gas analysis of as much importance when firing with natural gas as when using coal as fuel? Would employment of a CO_2 recorder be of benefit using natural gas for fuel with a steady load?

D. C. S.

Flue-gas analysis is as valuable for determining the completeness of combustion when burning gas as when coal is used as fuel. Steady load does not insure uniform conditions for combustion because gas and air pressure and atmospheric conditions are likely to vary, cracks may develop in furnace, or passages be left open which allow air to get inside of the setting. A CO_2 recorder affords means for determining whether operating conditions are being maintained.

Butt and Double Strap Joints

In double-riveted butt and strap joints of boilers, why is the narrower strap placed on the outside of the shell?

H. C.

Because the narrower strap is the one which has proper pitch of rivets for calking the joint.

Bleeder Turbine

What is a bleeder turbine?

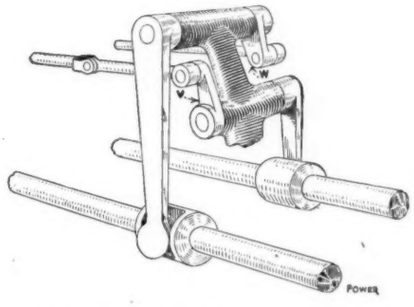
G. C. W.

A steam turbine in which provision is made for bleeding steam from one of the intermediate stages, for use in heating systems or manufacturing processes. (See POWER, Aug. 1, 1911, page 193.)

Valve Motion of Duplex Pump

In the valve motion of a duplex pump, why are the valve-rod rocker-arms, *V* and *W*, placed one above and the other below its rocker shaft? Could the pump be made to work with each valve-rod arm on the same side of its shaft?

N. H. J.



VALVE MOTION OF DUPLEX PUMP

In ordinary duplex pumps, the steam valve of one side is a duplicate of the other, and the valve-rod arms are placed on opposite sides in order to impart opposite direction of motion. Before reaching the end of its stroke one side of the pump has to move the valve of the other side, so the latter will move in opposite direction and the reversing action and operation of the pump are thus made continuous.

If the arms were the same, both sides of the pump would be as one side without means of reversing itself.

Chamfering of Tube Holes

Why should the edges of tube holes in a boiler-tube sheet be chamfered?

H. C.

The tube holes should be chamfered or rounded to a radius of about $\frac{1}{16}$ in. so that in expanding the tubes they will not be sheared by the edges of the tube holes.

Steam Used by An Ejector

What is the best method of determining the number of pounds of steam used per hour by an ejector?

C. A.

By taking the difference between the amount of water raised and the total discharged.

Testing Capacity of Safety Valve

What practical test can be made to determine whether the safety valve on a boiler is large enough?

E. M. S.

Making proper allowance for correctness of the steam gage, at a time when the gage indicates that the safe limit of pressure has been reached, the safety valve should be set so as to lift at that pressure. If the valve is large enough, then at any time after the proper limit of pressure has been reached, it should liberate steam so fast that the pressure cannot be raised above that point with the hottest fire that can be made and when all other escapes for steam are closed.

A test of the capacity of the valve should not be made unless the rate of firing can be increased very gradually and it is well established that the intensity of the fire is under complete control and provision is made for liberating the steam by an auxiliary outlet that can be opened for escape of steam in case the safety valve fails to work properly. For a boiler burning coal with natural draft the auxiliary outlet should have an area of not less than 1 sq.in. for each 2 sq.ft. of boiler grate surface.

Study Questions

This Week's Questions
Last Week's Answers

(41) While water is being drawn from a house tank at a uniform rate a pump running at 60 strokes per minute can fill the tank in 8 hr. and at 72 strokes per minute can fill it in 4 hr. With pumping stopped, how long would it take the full tank to empty?

(42) Under the same conditions as above, how long will it take to fill the tank again with the pump running at 84 strokes per minute?

(43) Two wires which have respective resistances of 5 and 7 ohms are joined in parallel so that a total current of 38 amperes flows through them. What current flows through each wire?

(44) What will be the rise of temperature of 1 lb. of water at 55 deg. F. when an ounce of copper at 212 deg. F. is immersed in it?

(45) Under a pull of 50 lb., how much will a steel tape stretch that is 100 ft. long, ½ in. wide and 0.02 in. thick if the modulus of elasticity of steel is 30,000,000?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(36) It is first necessary to find the tonnage capacity of the compressor, which is equal to the number of cubic feet of gas actually displaced per minute multiplied by the number of British thermal units of cooling effect available per cubic foot, divided by 200, the B.t.u. per minute equivalent to one ton of refrigeration per 24 hr. The actual displacement is equal to the volume swept through per minute by the piston, multiplied by the volumetric efficiency, 65 per cent. Therefore, the actual displacement would be

$$\frac{0.7854 \times 9^2 \times 13}{1728} \times 180 \times 0.65 = 56 \textit{ cu.ft.}$$

per minute. With a condenser pressure of 180 lb. and a suction pressure of 15 lb. the cooling effect per cubic foot of displacement will amount to 50 B.t.u. Then

$$\frac{56 \times 50}{200} = 14 \textit{ tons}$$

which is the tonnage capacity of the compressor. To find the rating of the engine necessary to drive the compressor, neglecting friction, multiply the tons capacity by 1.5,

$$14 \times 1.5 = 21 \textit{ hp.}$$

To include friction multiply by 1.7

$$14 \times 1.7 = 23.8 \textit{ hp.}$$

To coincide with commercial ratings it would then require a 25-hp. engine.

(37) With each forward and return stroke, taking 1 sec., the pump leaves 1 gal. in the standpipe. In 19 sec. the pipe has been filled to within 3 gal. of the top and on the next forward stroke water reaches the top, taking altogether then 19⅗ sec.

(38) Since the drop in voltage through transmission is 5 per cent. of the initial voltage and that at the building is 550 volts, the voltage at the power house must be

$$\frac{550}{0.95} = 578.95 \textit{ volts}$$

Hence the actual drop is 28.95 volts.

A load of two thousand 60-watt lamps and 50 hp. in motors is equivalent to

$$2000 \times 60 = 120{,}000 \textit{ watts (lamps)}$$
$$50 \times 746 = 37{,}300 \textit{ watts (motors)}$$

Total 157,300 *watts.*

Since the voltage is 550 the current will be

$$157{,}300 \div 550 = 286 \textit{ amp.}$$

From Ohm's law $I = \frac{E}{R}$ where I is the current, E the voltage drop and R the resistance, substituting,

$$286 = \frac{28.95}{R}$$

whence

$$R = \frac{28.95}{286} = 0.1012 \textit{ ohm}$$

is the resistance per mile of the line. The resistance per mile of wire would be $0.1012 \div 2 = 0.0506$ ohm.

(39) The fireman was trying to say that the injector had been fixed six days and was out of order 16 days. The calculation is a simple matter of addition and subtraction, using the plus sign for "after" and minus sign for "before" and adding a day for "tomorrow" and subtracting one for "yesterday," thus

$$+1 + 1 - 7 - 1 = -6$$

so it had been fixed six days *before*

$$+1 - 35 + 17 - 1 + 2 = -16$$

and it was out of order for 16 days *before* that. The calculation can be easily checked with the aid of a calendar.

(40) Any unit length of the pipe may be considered. Take it as 1 in. The pressure tending to burst such a ring 24 in. in diameter, separating it on any diameter, is ½ of $24 \times 1 \times 80 = 960$ lb. (See answer to study question No. 20, June 11, page 852.) This is resisted by a section of ⅜ × 1 in. = ⅜ sq.in. Then

$$\frac{960}{\frac{3}{8}} = 2560 \textit{ lb. per square inch}$$

While Colorado's coal production has decreased, its production in 1911 was over 10,000,000 tons.

One of the "Imperator's" immense rotors contains 50,000 blades, weighs 135 tons and can develop 22,000 hp.

Convention of National District Heating Association

On June 25 to 27 the National District Heating Association held its fourth annual convention at Detroit, Mich., with headquarters at the Cadillac Hotel. In every way, the meeting was successful, the attendance was 168, a number of good papers were presented and the entertainment provided was enjoyed by all.

At 10 a.m. on Tuesday the first session was opened with an address by Alexander Dow, of the Edison Illuminating Co., of Detroit. G. W. Wright, of Baltimore, responded. President Spencer's address will appear in an early issue.

Papers

Chairman A. P. Biggs read the report of the meter committee, which briefly described meters of interest to central heating companies. It included a discussion on system troubles which prevent correct metering and instructions on inspection, installing, reading and removal.

E. Darrow, in his paper on "Common Sources of Trouble in Customers' Installations," laid especial stress on lack of radiation and insufficient pipe capacity for zero-weather conditions. The determination of these factors is no longer left to architects and piping contractors. All plans are now submitted to the company's engineering department to be checked up before a heating contract is accepted. Too little attention is paid to radiator air valves. They must be sensitive to respond quickly to the cooling that takes place with an air accumulation. Every customer's installation must have at least one thermo-control valve set for 70 deg., usually placed near the entry of the service. For street mains, Mr. Darrow states, the three important requisites for construction are drainage of water accumulations from possible contact with the steam lines, provision for expansion and insulation.

In his paper on "Quality of Steam as Served by Central Stations to Their Customers," A. C. Shepherd gave his attention to tests of steam at different points along the lines of the Cleveland Electric Illuminating Co., the object of the tests being to determine the variation of quality at different points in the distributing system. Results and methods were given, using barrel and jet continuous condensing calorimeters.

R. D. DeWolf and J. L. Hecht, forming the committee on rates, presented a long report. Two sets of questions, one adapted to steam-heating systems and the other to hot-water systems, had been sent to 284 heating companies in the United States and Canada. Replies from 107 steam-heating and from 40 hot-water heating companies were received and tabulated in the report, the results indicating a wide range both in rates and methods followed in giving service. Contracts from

the various companies had also been requested and the contract clauses used by the various companies were summarized.

In "Analysis of Heating Rates," by A. E. Duram, the author did not attempt to determine any specific rate of charge, but rather suggested a method for such determination based upon the actual cost of the service rendered plus a fair percentage of profit. The great diversity in rates was shown from tabular data from various cities, and for rapid comparison a chart with a number of curves had been incorporated in the paper to show the net rate per square foot per season for all rates of consumption from 100 to 1200 lb. per square foot.

It is the duty of the committee on central-station records to collect data from all heating company members for the purpose of establishing standard forms of station and distribution records. This year and last it did not receive enough data to make complete records and it was decided to embody in the report a description of the system used by the Peoria Gas & Electric Co. This system is unusually complete and will undoubtedly furnish suggestions to other companies on which to start or improve their systems.

"Heating in Connection with Steam Turbines," by August H. Kruesi, presented some developments of the Curtis turbine bearing on the extraction of steam for district steam and for industrial heating, such as that required in bleaching, washing and dyeing. Features of operation in connection with such work were also discussed.

Edwin D. Dreyfus read a paper covering much the same subject, the turbine, however, being of the Parsons type. Both papers and most of the others herewith will be published in later issues.

J. F. Lewis described the power plant of the Murphy Power Co., Detroit, which was afterward inspected by many of the delegates. It carries a lighting and power, refrigerating and heating load which makes an excellent combination to maintain a uniform load factor.

A hot-water plant at Oak Park, Ill., was described by G. E. Chapman. This plant has 4800 hp. in boilers, 2000 hp. in turbines, 500 hp. in engines and approximately 12 miles of two-pipe heating mains.

"Sectional Steam Heating in Chicago" was by S. Morgan Bushnell. To supply steam for heating in the Loop district of Chicago, the Illinois Maintenance Co. was organized in 1900. Contrary to the usual practice, this company, with one or two exceptions, has installed no boilers of its own. It simply contracts to operate boiler plants already installed in buildings to furnish steam for heating and other purposes. Each plant, therefore, covers a very limited territory, one set of boilers being used in some cases for one building, although it is the intention to serve several adjacent buildings from each plant whenever contracts permit.

Each plant has its own operating force and all are supervised by a general engineer. The heavy investment for street mains is avoided, the transmission losses reduced, the operating efficiency increased over private plants and supplies in large quantities are obtained for less money. The company is giving satisfaction. Including steam for elevators, ice machinery, pumps, hot-water heaters, etc., it is supplying an equivalent of 800,000 sq.ft. of radiation.

In "Depreciation in Underground Distribution Systems," William Jennings, Harrisburg, Penn., gave a short account of his own experience in the Harrisburg Steam, Heat & Power Co. plant, originally installed in 1887. In making changes as late as 1911 it was found that there had been very little deterioration in the original system, which verified the author's claim that a properly installed system should last for 25 years.

A symposium on underground insulation by representatives of the various companies engaged in that line completed the list of papers.

Election of Officers

The officers for the ensuing year are president, R. D. DeWolf; first vice-president, J. F. Lewis; second vice-president, J. L. Hecht; third vice-president, E. Darrow; secretary-treasurer, D. L. Gaskill; executive committee, W. A. Wolls, H. R. Wetherell and A. D. Spencer, the retiring president.

Indianapolis as the next convention city was favored by the convention, but as requests had been received from numerous other cities, the final choice was left to the executive committee.

Entertainment

There was a theater party on Tuesday evening. On Wednesday the steamer "Pleasure" took the entire delegation, including the ladies, for a day's sail to the Flats and up the St. Clair River, ending at Belle Isle in the evening. Brief sessions were held on board to give time for a musical program and dancing. There was a buffet luncheon on the steamer and a crab and chicken dinner at the Casino, and in the evening a band concert on the island terminated a most enjoyable program. On Thursday at the close of the last session, touring cars took the delegates to "Delray."

Exhibits

These manufacturers exhibited heating specialties of all kinds, pipe insulation and meters: American District Steam Co., North Tonawanda, N. Y.; Builders Iron Foundry, Providence, R. I.; Central Station Steam Co., Detroit; Chamberlin Metal Weather Strip Co., Detroit; Jenkins Bros., New York; H. W. Johns-Manville, Milwaukee; Michigan Pipe Co., Bay City; the Richards-Wilson Pipe Covering Co., Grand Rapids; A. Wyckoff & Son, Elmira, N. Y.

Missouri N.A.S.E. Convention

On June 20 to 22, the Missouri State Association of the National Association of Stationary Engineers held its second annual convention at Joplin, and was well attended.

Allen McReynolds, of Zinc Belt Lodge No. 3, Joplin, was chairman at the opening session. After the Rev. Charles A. Weed's invocation, Mayor Jesse F. Osborne delivered the address of welcome. The response was made by National Secretary Fred Raven, as John W. Lane, editor of the *National Engineer*, was unavoidably absent. J. J. White, state president, spoke briefly of the work of the association during the year.

In the evening, at the exhibit hall, on the mezzanine floor of the Hotel Connor, F. E. Ransley, president of the Central States Exhibitors' Association, welcomed the visitors. An informal reception and luncheon were given by the exhibitors at which there were speaking and music.

At the Friday morning session Fred Raven, in one of his characteristic "keynote" addresses, spoke on the purposes of state conventions. State President White, in his report, told of his success in increasing the membership throughout the state. The secretary's report showed that there are six subordinate associations in good standing with a total membership of over 500 and that the prospects are bright for forming several more associations in the immediate future.

The committee on revising the constitution recommended several important changes. The *per capita* tax was raised from 10 to 15c. and provision was made to pay the delegates mileage with the funds left after all convention expenses have been met.

In the afternoon the ladies were taken for an automobile ride about the city. Four interesting lectures were given during the open session. George K. Elliott, chief chemist and metallurgist for the Lunkenheimer Co., spoke on "The Metallurgy of the Modern Steam Valve." He showed that the commercially successful valve must resist steam and strains, and the metal must be of such a nature as to cast easily and machine well. Pine copper is unsatisfactory as it is too soft to resist wire drawing, etc.; hence tin is added to produce the proper degree of hardness and strength. Zinc is used to properly finish the metal and make it steam-tight. Lead is afterward added to make the metal easy to machine.

The effects of sulphur, phosphorus and graphite on the qualities of iron were also discussed. Mr. Elliott illustrated his lecture with diagrams, tables and valves, showing the analysis and characteristics of various valve metals.

L. W. Pearce then read his paper, "An Engineer." This engineer is of the old school; he is little more than a starter and stopper; a handy man, with but little mental equipment to piece out his manual skill. He is rapidly passing, said Mr. Pearce, and but few of his type exist today. The progress made in steam engineering demands that a man be familiar with the theoretical part of the science of power generation and transmission. In conclusion, he said that manliness was the keystone of success in steam engineering and, coupled with good habits, this quality would give anyone a long start.

Herbert Taylor, of the Du Pont Powder Co., spoke briefly on "New Application of Concentrated Energy, 'Dynamite.'" Crop yields have been increased over 300 per cent. by thoroughly dynamiting a field to a depth of from 3 to 5 ft. By thus loosening the soil, more moisture is absorbed and retained and the plants are better nourished.

A. Herron, of St. Louis, read F. H. Munsberg's paper, "The Educational Feature of the N. A. S. E. and the Benefits Derived from Attending Meetings," as Mr. Munsberg was unable to be present.

In the evening the delegates and visitors were entertained at Lakeside Park, an amusement resort about 10 miles from Joplin, where the Essany Film Manufacturing Co., Chicago, gave a special moving-picture show.

At the executive meeting on Saturday morning, Springfield was selected as the next meeting-place. These newly elected officers were installed by National Secretary Raven: President, F. H. Munsberg, St. Louis; vice-president, D. M. Trimble, Kansas City; secretary, J. H. Chestnutt, Springfield, reëlected; treasurer, A. P. Maeder, Joplin; conductor, O. M. Evans, Springfield; doorkeeper, W. E. Rogers, St. Louis, and trustees, D. Spoerndli, St. Joseph; F. B. Daggett, St. Joseph, and Charles Parkinson, St. Louis. A joint meeting of the delegates and exhibitors was then held, at which members of both associations spoke.

On Saturday afternoon a trolley trip was taken to the American Zinc, Lead & Smelting Co.'s mining plant, where the power plant was inspected and the process of recovering the lead and zinc ores was observed. The power plant and employees' club house of the Southwestern Missouri Railway Co. were then visited, after which the party returned to Joplin.

The evening was spent in visiting the numerous and interesting exhibits. The exhibitors, members of the Central States Exhibitors' Association, were: American Steam Gauge & Valve Co.; V. D. Anderson Co.; Crandall Packing Co.; Dearborn Drug & Chemical Works; Garlock Packing Co.; Greene, Tweed & Co.; Hawk-Eye Compound Co.; Hills, McCanna Co.; Home Rubber Co.; Jenkins Brothers; H. W. Johns-Manville Co.; Keystone Lubricating Co.; *National Engineer;* Osborne Valve & Joint Co.; William Powell & Co.; POWER; Schaeffer & Budenberg Co.; C. E. Squires Co.; United States Graphite Co.; Joplin Supply Co.; Rogers Foundry & Manufacturing Co.; the Missouri Fidelity & Casualty Co.; Landreth Machinery Co.; Waters-Pierce Co.; DuPont Powder Co.; Nowlin Machinery Co.; George B. Carpenter & Co.

Connecticut N. A. S. E. Convention

On June 28 and 29, the seventeeth annual convention of the Connecticut State Association of the N. A. S. E. was held at Bridgeport, the headquarters being the Stratfield Hotel. About 50 delegates were present from the 10 local branches, besides many prominent visitors from New York, New Jersey and Massachusetts. The meetings were held in Colonial Hall, which was tastefully arranged for the mechanical exhibit, fully 50 firms occupying booths.

Friday morning at 10:30, the convention was formally opened by State President F. Chapman. Mayor C. B. Wilson welcomed the engineers and their guests to the city, National President Edward Kearney responding. F. J. Mulcahy, vice-president and general manager of the Crane Valve Co., Bridgeport, addressed the convention at length. The several committees were then appointed, and the convention adjourned until Saturday at 9 a.m.

On Saturday morning, business was quickly disposed of. The reports showed a big increase in membership and that the organization was in a good financial condition. The officers for the ensuing year are Thomas J. Phillips, president; Bernard Reynolds, vice-president; Charles M. Dowd, secretary-treasurer; Herman Offinger, conductor; Jacob Miller, doorkeeper; William Holt, George P. Thomas and Roscoe Granger, trustees. F. D. Hastings was reëlected state deputy.

A meeting of the supplymen was held on Friday afternoon, President F. S. Bulkley in the chair, and a new organization, the Connecticut Supplymen's Association, was formed, for which a charter will be asked. Its aim is to assist the local committees in arranging for exhibits and in entertaining the delegates and guests. The following officers were elected: F. P. Upson, president; E. S. Birge, vice-president; W. L. Larash, secretary-treasurer; John H. Foote, F. S. Bulkley, R. H. Stiles, J. B. Harrington and C. H. Tower, executive committee.

A visit was made to the plant of the Crane Valve Co. on Friday afternoon and during the evening, in the exhibition hall, Past National President Herbert E. Stone talked on "The Relationship of the Engineer to the Supplyman." C. Cooley, superintendent of power of the New York, New Haven & Hartford R.R., gave an address on "Recent Installations of Power Plant Equipment." H. J. Maynard, of the General Electric Co., read a paper on "Artificial Illumination." Interspersed between the speeches were songs by Frank Corbett, of the Consolidated Valve Co., and William Murray, of Jenkins Brothers. Frank J. Quinlan, chairman of the local committed, presided.

On Friday night the delegates were entertained by the supplymen with a smoker in the rathskeller of the Stratfield Hotel. On Saturday afternoon the company enjoyed a pleasant sail on Long Island Sound. It was decided to hold the next annual convention at Meriden.

EXHIBITORS

The exhibitors were the American Steam & Valve Manufacturing Co., Ashton Valve Co., Armstrong Manufacturing Co., Albany Lubricating Co., Bridgeport Testing & Laboratory Co., Crandall Packing Co., Connecticut Metal Boiler Compound Co., Crocker-Wheeler Co., C. A. Claflin Co., Crane Valve Co., Durabla Manufacturing Co., Dearborn Drug & Chemical Works, M. T. Davidson Co., General Electric Co., Garlock Packing Co., Greene, Tweed & Co., Harrison Safety Boiler Co., A. W. Harris Oil Co., Handy Manufacturing Co., Hartford Mill Supply Co., T. Hawley & Co., Home Rubber Co., Hawley Hardware Co., W. J. Hyland Manufacturing Co., Heine Safety Water Tube Boiler Co., The Ironworks Co., Jenkins Brothers, H. W. Johns-Manville Co., Keystone Grease Co., Lunkenheimer Co., Lyon & Grumman, E. H. Ludeman, Estate Edward R. Ladew, Mason Regulator Co., McLeod & Henry Co., C. S. Mersick, N. Y. Belting & Packing Co., Nightingale & Childs, *National Engineer, New England Engineer,* National Flue Cleaner Co., Otis Elevator Co., Perfection Grate Co., Patterson Allen Co., POWER, Peerless Rubber Manufacturing Co., Quaker City Rubber Co., Roto Co., J. R. Robinson, Smooth-On Manufacturing Co., *Southern Engineer,* Standard Regulator Co., Schaeffer & Budenberg Manufacturing Co., Skinner Engine Co., Traveler's Insurance Co., George H. Thacher Co., and W. R. Winn.

PERSONAL

Dr. Leonard A. Levy, head of the chemistry and research department of Alexander Wright & Co., Ltd., has just been awarded the degree of Doctor of Science by the University of London.